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USAAVLABS TECHNICAL REPORT 67-2

STUDY AND DESIGN
OF
ARMORED AIRCREW CRASH SURVIVAL SEAT

By

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March 1967

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-263(T)

HAYES INTERNATIONAL CORPORATION
BIRMINGHAM, ALABAMA

DEPARTMENT OF THE ARMY
U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS. VIRGINIA 23604

This report has been prepared by Hayes International Corporation under the terms of Contract DA 44-177-AMC-263(T). The technical objective of the contract was to design an armored crew seat incorporating the results and recommendations of previous research efforts involving the dynamic testing of four armored, crashworthy, experimental aircrew seats.

The results of this research effort indicated that it was feasible to embark on a fabrication and test program of experimental crew seats in order to justify the design criteria and the energy absorption systems. The results will be utilized in the design of passive defense systems for future Army aircraft.

Views expressed in this report have not been reviewed or approved by the Department of the Army.

Task 1F121401A15003
Contract DA 44-177-AMC-263(T)
USAAVLABS Technical Report 67-2
March 1967

STUDY AND DESIGN
OF
ARMORED AIRCREW CRASH SURVIVAL·SEAT

Hayes International Corporation Report No. 1311

by

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Prepared by

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Birmingham, Alabama

for
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FORT EUSTIS, VIRGINIA

SUMMARY

The primary objective of this program was to develop the design of an improved aircrew armored crash survival seat. The U.S. Army, through previous contractual efforts, designed and developed four armored aircrew seats having crash load attenuation features. Each design was unique in concept, geometry, material, energy absorption and other factors. A comprehensive analysis and evaluation were made of these designs as a basis for developing an improved design.

The dynamic tests did not yield a significant amount of useful data in the determination of the load attenuation characteristics of the seats. The energy absorbing devices did not function as designed. This was due to premature failures and to the inadequacy of the occupant restraint system to maintain the occupant in a position to load the seat in accordance with the design concept.

Design criteria for strength and deformation characteristics are based on recommendations by Aviation Safety Engineering and Research (AvSER) in their Technical Report 65-14, entitled "Proposed Military Standard For Improved Crew/Passenger Survival Seats and Body Retention Systems". Basically, these requirements were: The initial seat collapse strength under longitudinal loading shall exceed 35g, based on an occupant weight of 200 pounds; It must be capable of vertical deformation of not less than 6 inches while maintaining a vertical load of 17g based on an occupant weight of 80 percent of the weight of a 200-pound occupant; It shall exhibit static strength such that initial collapse occurs under lateral loading of 4,000 pounds.

A seat design was developed that will meet or exceed the specified requirements. It is forward facing and designed for installation in the UH-1B aircraft. It consists of a seat bucket, fabricated of aluminum alloy sheet, supported by a tubular steel framework from floor tracks. Load attenuation is accomplished by a Hayes-developed energy absorbing device utilizing controlled bending of steel rods. The seat bucket is allowed to move relative to the support for energy absorbing stroke in the vertical and lateral directions. The complete seat moves relative to the floor for the stroke in the longitudinal direction. The load attenuation devices can be adjusted or easily replaced for various occupant-armor weight configurations. Vertical adjustment is accomplished with an electric actuator.

The restraint harness designed for minimized elongation as well as strength consists of a lap belt, shoulder straps, and a crotch strap. A dual inertia reel is mounted on top of the seat back. All straps terminate at a single point at which all can be released immediately.

Ballistic protection is provided by a shell of ceramic-fiber glass composite armor attached to the seat bucket. The armor was developed by Cincinnati Testing Laboratories and is somewhat lighter in weight than present armor. Protection is provided against 7.62mm APM-61 ammunition fired at 100 yards range at 15° obliquity.

<u>FOREWORD</u>

A program was conducted by Hayes International Corporation to develop the design of an improved armored aircrew crash survival seat for use in helicopters. This included the analysis and evaluation of four different armored seats having energy absorbing capabilities that were subjected to dynamic tests. The program was accomplished under Contract DA44-177-AMC-263(T) for the U.S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia. The contract was initiated in June 1965. It was accomplished in three phases as follows: Phase I, Analysis of the Four Seat Designs; Phase II, Development of a Concept of An Improved Seat; and Phase III, Detail Design.

The program was conducted under USAAVLABS direction of Mr. F. P. McCourt, Chief, Safety and Survivability Division. Mr. R. Fama, Project Engineer, was designated as authorized representative of the Contracing Officer, Mr. R. P. McKinnon.

Principal Hayes engineers were: Mr. L. R. Anderson, Project Engineer; Mr. W. T. Holmes, Mr. G. R. Grimes and Mr. O. A. Rogers, Analysis; Mr. L. B. Wheeler and Mr. B. L. Lewis, Designers.

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TABLES

SYMBOLS

A	area
BL	buttock line
C	crotch strap load
c	distance from neutral axis to extreme fibers
c.g.	center of gravity
d	diameter
E	modulus of elasticity
F	allowable stress
f	actual stress
I	moment of inertia
k	section factor
L	length, lap belt load
M	bending moment
M.S.	margin of safety
P	load
R	reaction force; ratio of actual to calculated loads; ratio of actual to calculated stresses
r	radius
S	section modulus; shoulder harness load
T	torque
U	energy
V	shearing force
WL	water line
$X' - X'$	reference axis
$Y - Y$	reference axis
$Z' - Z'$	reference axis

₵	center line
⊕	center of mass
δ	deflection; plastic correction term
ε	fiber strain
μ	friction coefficient
σ	theoretical stress

SUBSCRIPTS

br	bearing
c	compression
cc	crippling
b	bending
s	shear
st	torsional shear
t	tensile
u	ultimate

INTRODUCTION

It has been repeatedly shown from aircraft crash investigations that a higher degree of energy absorption is needed in present design of aircrew seats. This has been revealed in cases where impact velocities and resulting decelerative forces were such that with reasonable seat restraint and energy dissipation, crew survival would have been possible. These investigations have led to a greater awareness of the need for a reappraisal of design criteria and design concepts in obtaining crashworthiness of aircrew seats.

The U.S. Army, through contractual effort, has designed and tested four aircrew crash survival seat concepts. These seats were designed to incorporate armor materials and energy absorption techniques to increase the occupants' probability of survival under crash impact and ballistic threat conditions. Each of the seats is unique in geometry, material, energy absorption, and other factors. They were subjected to dynamic tests to evaluate their crashworthiness properties and to develop deceleration time histories, load deformation curves, and other pertinent data.

A comprehensive study and evaluation of these designs were accomplished. This included the design philosophy, design criteria, detail design, energy absorption characteristics and the results of the dynamic tests relative to the design objective. A review of the design criteria, a discussion of each of the designs (including conclusions), and a comparative evaluation are included in this report.

The primary objective of this program was to develop an improved aircrew seat design. The original contract requirements were for a seat having load attenuation capabilities in the vertical direction only and designed for rather high loads. Preliminary design study indicated that a seat meeting these requirements would be quite heavy and would not provide the necessary crash protection. The design requirements were revised to incorporate recommendations by Aviation Safety Engineering and Research (AvSER), a Division of Flight Safety Foundation, Inc. These criteria are published in AvSER Technical Report 65-14. They require load attenuation in longitudinal, lateral, and vertical directions. Design load requirements were reduced from those originally specified in the contract. A design of an armored crashworthy seat was developed that will meet all of the specified requirements. This report contains a summary of all work accomplished on the contract, including results of evaluation and design study. It also contains complete load and structural analysis of the seat design.

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ANALYSIS AND EVALUATION OF SEAT DESIGNS

The primary objective of this program was to design an improved aircrew armored crash survival seat. The four seat designs previously developed and subjected to dynamic tests were to serve as useful basic data for this design development. A thorough analysis and evaluation of these designs were accomplished. This study included an evaluation of the design criteria, design philosophy, detail design, materials, energy absorbing characteristics, and results of the dynamic tests relative to the design objective. Although none of the four seats were considered to be optimum in any respect, each design is unique in geometry, material, energy absorption and other factors. Data available for this study included engineering drawings, stress analyses, results of the dynamic tests, high speed movies of the tests, and one each of the seats that had been subjected to the dynamic tests.

Most of the conclusions relative to the seats were based on design and analytical data. The dynamic tests did not yield a significant amount of useful data. This was due to premature failures and to the inadequacy of the occupant restraint system in preventing the energy absorbing devices from functioning as designed. The following paragraphs include a review of the design criteria and a description and discussion of each of the designs, including conclusions and a comparative evaluation.

DESIGN CRITERIA

The design criteria used for the four seats that were evaluated differed in several respects. The seats were designed for approximately the same, but not identical, structural criteria. Variations existed in magnitude of load factors and also in the manner in which they were combined. The basic guidelines for other requirements apparently varied. Three of the seats were designed for floor mounting. One was designed to be supported from an airframe bulkhead, while another approach was to develop a universal seat that would be adaptable to a number of Army aircraft.

All of the seats have vertical load attenuation capabilities that were designed for essentially the same criteria of limiting vertical accelerations in the pelvic region of the occupant to a nominal value of -20g. In general, each of the seats, its support system, and the occupant restraint system, individually and in combination, were designed to have sufficient strength to withstand the loads from longitudinal deceleration of 25g for 0.20 second and 45g for 0.10 second in the pelvic region of a suitable anthropormorphic dummy having a weight and mass distribution of that of the 95th percentile man. The weight of a 95th percentile man is 200 pounds.

A summary of the design accelerations is given in Table I. These data are as presented in USAAVLABS Technical Reports 64-73, 65-2, and 65-7. Also given in the table are preliminary type analytical data on the CH-34 seat design.

Although the seats were designed by different load criteria, the series of

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TABLE I

DESIGN ACCELERATIONS

Seat	Condition	Individually Applied			Combined		
		Long. (g's)	Lat. (g's)	Vert. (g's)	Long. (g's)	Lat. (g's)	Vert. (g's)
UH-1B	Forward	45	-	-	-	-	-
	Side	-	45	-	-	-	-
	Vertical	-	-	20±5	-	-	-
	Forward-Side-Vertical	-	-	-	25	25	25
Universal	Forward	45	-	-	-	-	-
	Side	-	10.5	-	-	-	-
	Vertical	-	-	21.4	-	-	-
	Forward-Side-Vertical	-	-	-	45	10.5	21.4
	Forward-Vertical	-	-	-	45	-	21.4
	Forward-Side	-	-	-	45	10.5	-
	Side-Vertical	-	-	-	-	10.5	21.4
CH-34	Forward	45	-	-	-	-	-
	Side	-	45	-	-	-	-
	Vertical	-	-	20	-	-	-
	Forward-Side-Vertical	-	-	-	45	45	20
	Forward-Vertical	-	-	-	45	-	20
	Forward-Side	-	-	-	45	45	-
	Side-Vertical	-	-	-	-	45	20
CH-47	Maneuver	-	-	±5	-	-	-
	Forward	45	-	-	-	-	-
	Side	-	45	-	-	-	-
	Vertical	-	-	25	-	-	-

tests were essentially the same for all seats.

DESCRIPTION OF SEATS

UH-1B

This seat was designed for installation in the UH-1 aircraft and was adapted from another UH-1B crew seat. It consists of a seat bucket fabricated of sheet aluminum alloy. A triangular frame made of steel tubing and fittings provides support for the seat bucket and is attached to floor-mounted seat tracks. Four aluminum slide fittings attached to the back and sides of the bucket provide restraint for the bucket to the frame. These fittings are interconnected horizontally by steel tubes and slide vertically on the aft member of the steel frame for load attenuation travel. The seat is attached to the airplane through two tracks connected between the fore and aft vertical members of the seat frame. Fore and aft adjustment is accomplished by positioning spring-loaded pins into holes in the seat floor track. Vertical adjustment was not provided in this design. Figure 1 is a photograph of the seat.

Vertical crash forces are attenuated by a series of up to eight cable-sheave energy absorbers which are mounted between the bucket fitting interconnect tubes and the horizontal member of the seat frame. This device functions by progressive failing of lips of the sheave which are pressed over a steel cable circling the periphery of the sheave. A vertical travel of 7.9 inches is available for load attenuation.

A lap belt, a crotch strap and a shoulder harness terminating in a single inertia reel comprise the restraint system. Instantaneous release is provided by one lever on the belt buckle. The complete restraint system is mounted on the seat bucket to prevent loosening should the seat stroke downward.

Ballistic protection is provided for .30-caliber ammunition by armor fabricated from a hard-faced ceramic armor utilizing 6-inch-square tile. The seat bucket is adapted to incorporate armor protection across the back, bottom, and front. Additional armor for side protection was to be attached to the aircraft entrance door. A readily removable chest protector provides front armor. It is attached to the seat bucket by pivoted arms to allow the necessary pilot movement and to permit the armor to swing forward away from the pilot's chin in the event of a crash.

Universal Seat

The seat design consists of a seat bucket fabricated basically of aluminum alloy but utilizing armor materials for some structural elements. The seat support structure is also fabricated of aluminum sheet and machined parts. Two vertically oriented tracks on the support structure provide support for the seat bucket and the necessary travel for energy absorption. The seat bucket is attached to the tracks through rollers on the inside of the support frame and at the center of the back of the support frame for restraint

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of the bucket to the frame. The seat is attached to the airframe through tracks mounted at each end of the side support members. Fore and aft adjustments are provided by positioning spring-loaded pins into holes in the floor-mounted seat tracks. Vertical adjustment is provided by shear lugs on the bucket which latch into a notched guide on the seat support. Stroke length for energy absorption is reduced as the seat is adjusted to a lower position. A photograph of the Universal seat is shown in Figure 2.

Attenuation of vertical crash accelerations is accomplished by absorbing energy by bending and rebending steel straps suspended parallel to the back of the seat. Bending is accomplished by forcing the straps to bend around a five-pin carriage assembly attached to the seat bucket.

The occupant restraint system consists of a shoulder harness, a lap belt, and inverted "vee" belt type thigh straps. The shoulder harness and the "vee" belt terminate in metallic loops that fit over a quick-release lap belt fastener. A single inertia reel mounted on the top of the seat bucket back is used in the shoulder harness.

Ballistic protection is provided by armor material mounted on the side, bottom, and back of the seat bucket. The bottom and back armor material is used as structural elements of the seat bucket. In addition, a shoulder panel is cantilevered from the upper right corner of the seat bucket, and a front torso shield is provided which completely envelops the upper chest area. Both the shoulder panel and the torso shield are hinged to allow easy egress. The torso shield is attached to the seat bucket through a support linkage and a nylon strap which encircles the shield and is attached to the support linkage. The armor material is a hard-faced composite, made from ceramic tile with laminated backing.

CH-34

This seat was designed for installation in the CH-34 aircraft. It is supported on two vertical tracks mounted on a bulkhead behind the seat. The seat bucket is of all-aluminum alloy construction utilizing panels of honeycomb sandwich, sheet and machined parts. The back of the bucket consists of a honeycomb sandwich panel attached to two vertical members, one on each side. These vertical members are channel shaped and engage the fixed tracks mounted on the bulkhead. The bottom of the seat bucket is a honeycomb panel supported on the sides by aluminum plates attached to the vertical members. Vertical crash forces are attenuated by crushing an aluminum honeycomb material inserted in the seat support tracks. Figure 3 is a photograph of the seat.

Occupant restraint is provided by a seat belt, shoulder harness, and crotch strap connected at one point on the seat belt to a post-type fastener on the lap belt and secured by a spring loaded safety pin.

Ballistic protection is provided by the armor panels used as structural members in the sides and back of the seat and by panels mounted on the bottom and front lip of the seat pan. A chest protector is provided which

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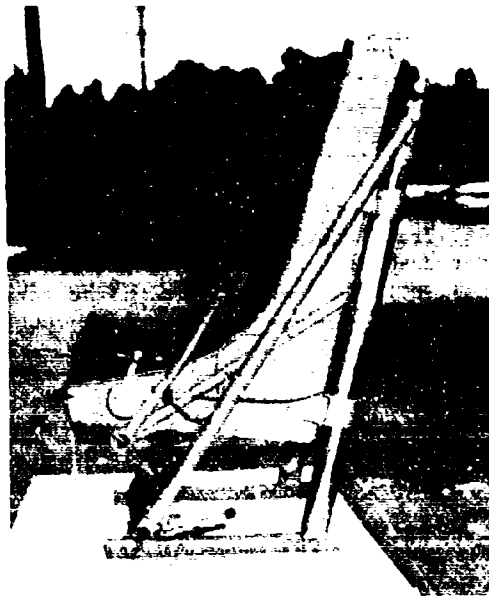
Figure 1. UH-1B Seat.

Figure 2. Universal Seat.

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Figure 3. CH-34 Seat.

Figure 4. CH-47 Seat.

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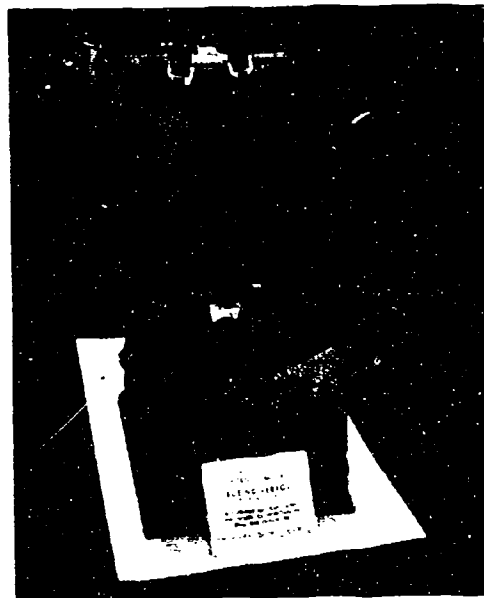
is mounted in slots on each of the side armor panels. The chest protector is hinged at the top to allow motion of the pilot during flight operations.

CH-47

This seat was designed for installation in the CH-47A helicopter. It is unique in that it uses all of the ballistic protective armor as structure. It consists of a seat bucket formed by fastening hard-faced ceramic armor to a light aluminum skeletal frame. A series of aluminum Trussgrid* honeycomb type blocks support the seat bucket from the floor. The seat is attached to a floor plate through the Trussgrid honeycomb blocks, and the floor plate is attached to the aircraft floor. These blocks have glass cloth laminated facings to which hinges are attached. The blocks are hinged at the bottom of the seat bucket and at the floor. The seat is constrained in the longitudinal direction by a screw jack at the bottom and by a nylon strap and inertia reel at the top. The free end of the strap is attached to a bulkhead behind the seat.

Vertical and horizontal adjustments are accomplished by moving the screw jack. As the screw jack is actuated, the support blocks pivot around one edge, providing vertical and horizontal adjustment simultaneously. A photograph of the seat is shown in Figure 4.

Vertical and lateral crash forces are attenuated by crushing the honeycomb support blocks and a block of honeycomb which is attached between these blocks and the bottom of the molded armor seat bucket.

Ballistic protection is provided by the armor used in the seat bucket. In addition, a chest protector and shoulder shield are provided. The shoulder shield is hinged at the top right corner of the seat bucket. The chest protector is supported vertically by a single fitting which connects at the center front lip of the seat pan. Longitudinal restraint of the chest protector is provided by the shoulder harness.

Occupant restraint is provided by a seat belt and shoulder harness. Since the chest protection is installed inside the shoulder harness and has a load reaction point at its bottom support, it also contributes, to a degree, to the longitudinal restraint.

ENERGY ABSORPTION

Each of the four seats utilized a different and unique device for vertical load attenuation. None of the seats tested fulfilled all of the design objectives. The energy absorption devices functioned in the intended manner on only one of the seats in one test. Therefore, their energy absorbing and load attenuation capabilities were not demonstrated. Excessive elongation in the restraint system allowed the dummy to assume a position well forward. This resulted in a load distribution on the seat that differed

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*General Grid Corporation, Edgewood Arsenal, Maryland.

from the design analysis, and the energy absorption device was not loaded as intended. Except for one of the devices, it is believed that they would have functioned as intended if the restraint system had held the dummy in the proper position.

Following is a brief description of each of the concepts for energy absorption and comments on their characteristics, design, and feasibility.

UH-1B

Energy absorption in the UH-1B seat is accomplished by a series of cable-sheave type devices connected between horizontal members on the seat bucket and support frame. These devices are designated "Attenuator Assembly, Mark III, P/N B-9923-3". The attenuator assembly consists of an aluminum sheave with a steel cable imbedded in a groove around the outer periphery. When a load of predetermined value is applied to the cable, energy is absorbed by progressive failure of the lips of the sheave which are pressed over the cable. The sheave rotates about its center, which is fixed to the support structure. Figure 5 is a sketch of this device.
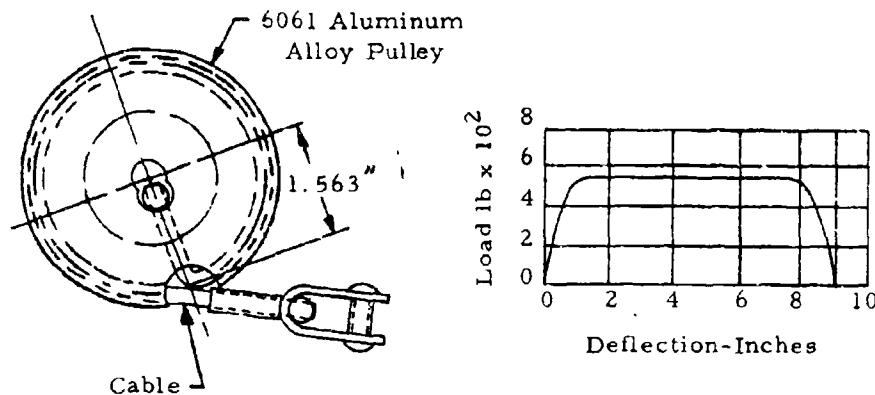
Figure 5. Attenuator Assembly, Mark III Cable-Sheave Energy Absorber.

Data in the UH-1B stress analysis, Reference 10, indicate that the device will sustain a constant load of 550 pounds over a stroke length of nine inches as shown in Figure 5. Available deflection computed from drawing data is 7.9 inches.

Provisions were made for the use of up to eight of the attenuators and eight were used in the tests. Loads from sliding friction were accounted for in arriving at loads on the attenuator. The magnitude of the loads due to friction is rather difficult to compute and could have contributed quite heavily in preventing the energy absorber from operating. From an examination of the sliding components, it is concluded that additional effort

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should have been made in the design to reduce the friction.

Although the energy absorbing capability was not demonstrated in the tests, the concept and the device are considered good.

Universal Seat

This design uses a metal bending device installed parallel to the seat back connecting the seat bucket to the support frame. It functions by bending and rebending metal straps around a set of pins as shown in Figure 6. The straps are fixed to the support structure, and the five-pin movable carriage is attached to the seat bucket. The device was sized for this specific application and is rated at 5000 pounds ±5 percent to limit the loads on the occupant to 21.5g. The seat bucket moves on rollers, resulting in relatively low friction under forward loadings but with sliding friction for lateral loading.
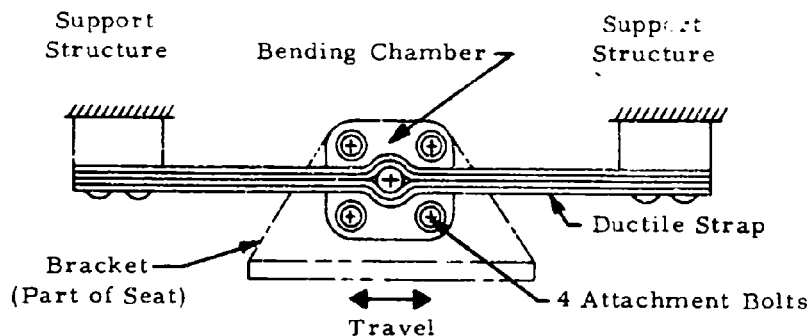
Figure 6. Van Zelm Metal Bending Energy Absorber.

In one of the tests, this device started to actuate, but the complete seat became disengaged from the track. It is reasoned that the seat experienced accelerations of more than 20g only instantaneously, and the question remains as to whether it would have limited the acceleration on the dummy to -21.5g, the design value.

This concept and design for vertical energy absorption are considered to be good. Additional drop tests on the seat were conducted under this contract. The results of these tests are summarized in another section of this report.

CH-34

Crushing of aluminum foil honeycomb material attenuates the vertical loads in the CH-34 seat. The honeycomb is enclosed in two vertically oriented channels on the aft side of the seat bucket. It is surrounded by the channel on three sides and partially on the fourth. This "energy strut" engages the fixed bulkhead-mounted track in such a manner that the fixed track compresses the honeycomb parallel to the longitudinal axis of the

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cells as the seat moves downward when a predetermined vertical load is attained. In order to prevent normal operational and vibratory loads from progressively crushing the edge of the honeycomb, the seat is supported by two shear pins. Compression of the energy strut is activated when these two pins are sheared. A sketch of the concept is shown in Figure 7.

Typical compressive load-deformation characteristics of aluminum honeycomb are shown in Figure 7. Test data show that there is a fairly steady increase in load until the yield point is reached, followed by an abrupt decrease in load after the maximum yield strength is passed. Crushing can be continued until about 25 percent of the original thickness of the honeycomb remains.

The CH-34 energy strut has a free length of 18.75 inches. The honeycomb is precrushed to an installed length of 18.5 inches to eliminate the high peak load as indicated in Figure 7. The designer conservatively assumed an efficiency of 66 to 70 percent to give an effective stroke for energy absorption of 12.1 to 12.9 inches. The shear pins are designed to fail and the energy strut becomes operative at 18.2 to 19.4g with a 200-pound occupant in the armored seat. The specified design limitation was 20g ±5g. The system did not function as intended in the dynamic tests.

The concept of crushing honeycomb material for energy absorption for aircraft seats is considered to be feasible and good. However, it is concluded that the system did not function as desired because of a deficiency in the detail design. (Friction in the track supports apparently was neglected or inadequately accounted for in the determination of loads on the shear pins and energy strut.) The strut should compress as intended under vertical load only. Actually the moment due to the cantilevered seat bucket increases fore and aft loads on the support. If a high friction coefficient exists in the track system, the vertical loads on the shear pin and energy strut would be reduced significantly. The components used in the seat were 7075-T6 aluminum alloy, and the sliding surfaces were coated with zinc chromate primer. It is reasonable to assume that the coefficient of friction was quite high. For a load condition as tested, wherein there was a high forward acceleration combined with the vertical acceleration, the loads on the energy strut were greatly reduced. A load analysis indicates that the friction loads were of sufficient magnitude to have prevented the system from operating in the tests. Friction must be greatly reduced and controlled to make this concept feasible.

CH-47

This design utilizes a unique system for energy absorption. Energy is absorbed by the crushing deformation of blocks of Trussgrid, a honeycomb type material. The bottom of the seat bucket is made of Trussgrid, but the major energy absorbing elements are blocks of Trussgrid between the seat bucket and the floor. These blocks rotate for combined vertical and horizontal adjustment. They are loaded diagonally and form a parallel link system driven by a floor-mounted electric linear actuator. An automatic seat-mounted inertia reel and strap at the top of the seat back are used to re-

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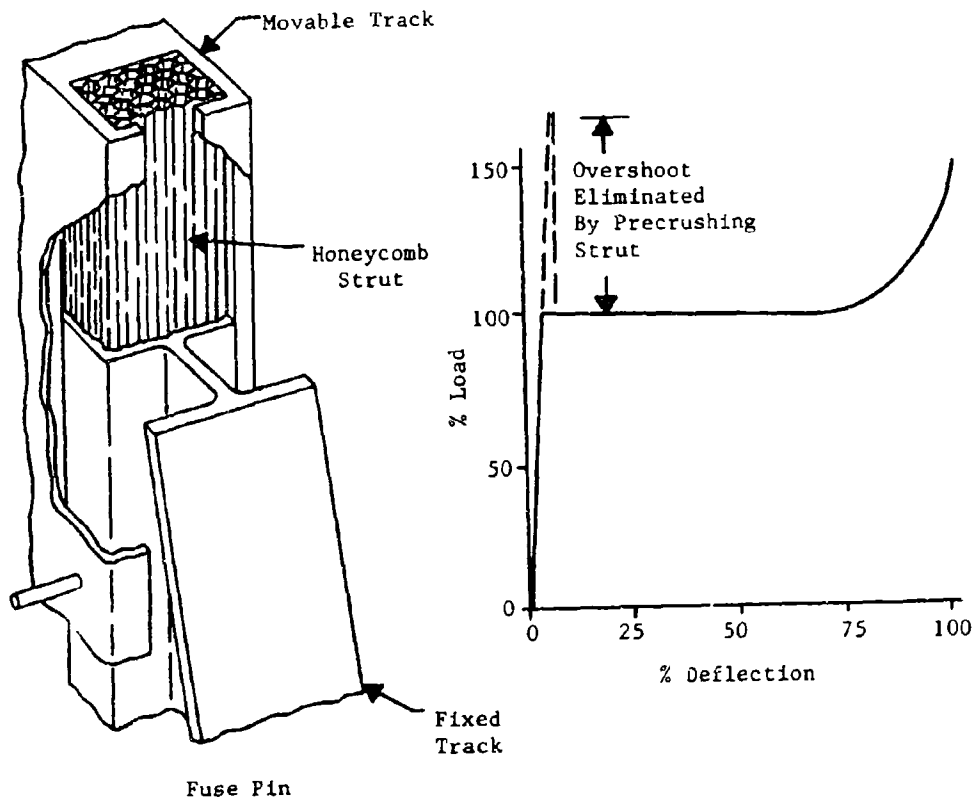
Figure 7. CH-34 Energy Absorbing System.
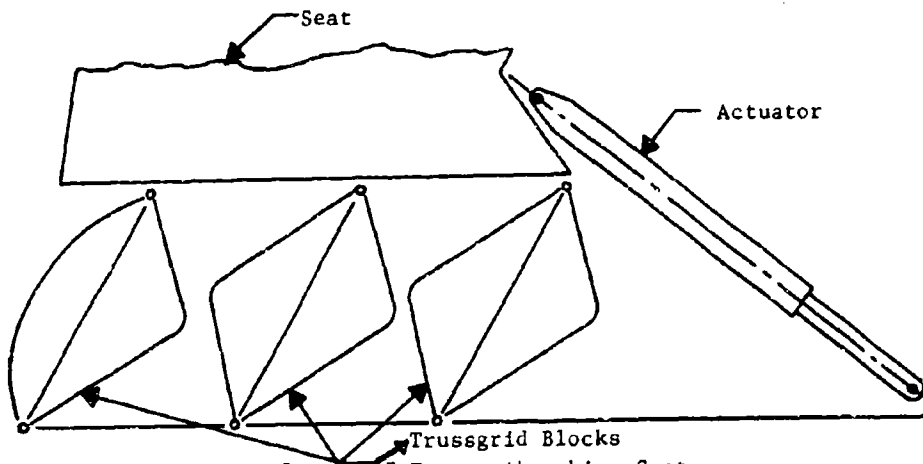
Figure 8. CH-47 Energy Absorbing System

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strain the seat and occupant in forward and side crash accelerations.

The design goal was for vertical loads to be attenuated to 20g ±5g continuously maintained in the pelvic region through crushable blocks and through a crushable pedestal attached directly to the seat pan armor.

The adjustable blocks are arranged to vary the load-limiting action to adjust to man weights from 135.9 pounds to 199.7 pounds. This is automatically obtained when the occupant adjusts the seat for his optimum height and reach. This system assumes that the 5th percentile man at 135.9 pounds will adjust his seat to the highest position, and that the 95th percentile man at 199.7 pounds will adjust to the lowest position. Also, it assumes that those men who fall between this range will have a height and weight proportional to their percentile. This system does not make allowance for the extremes such as a 5th percentile height man weighing 199.7 pounds or a 95th percentile height man weighing 135.9 pounds. Therefore, the seat occupant must consider his weight when making a seat adjustment to obtain optimum vertical crash protection. Since this may not always be done, a non-optimum load limit may sometimes occur.

The longitudinal and lateral design loads of 25g for 0.20 second and 45g for 0.10 second are accommodated by allowing the crushable seat pedestal and adjustable blocks to deform. This allows the restraining members to act in tension. In a forward crash, restraint is accomplished by distributing the loads in three locations:

- At the seat top-center, by a strap which is attached to the bulkhead behind the seat. The strap acts through an inertia-sensitive reel on the back of the seat.

- At the back of the seat, at the intersection of the bucket and pan, by tension on the adjustable screw jack. The jack is attached to the seat base at the floor bulkhead intersection.

- At the adjustable blocks, by partial crushing of the combined pedestal and support structure. This provides some attenuation at the load onset, and it provides position retention thereafter.

The side load retention assumes that a considerable deformation of the combined crushable base will take place before the main restraint members (strap and screw jack) take up the load. The amount of travel to either side will vary from 5 to 8 inches, depending on the initial seat adjustment.

This concept of energy absorption is considered to be good and has merit for crew seat application. The kinematics and kinetics of the system are quite complicated and require intricate analysis. Many variables or parameters enter into the analysis, and the validity of the analysis depends on proper representation of these parameters. It is believed that more know-

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ledge of some of the variables is necessary to design the system properly. The load-deflection properties of the Trussgrid blocks for the various load directions should be determined by tests.

The crash performance of the seat in the tests was limited by failures of the restraint system. There was evidence of energy absorber action, but an evaluation of the crash load mitigation properties cannot be made from the test results.

RESTRAINT-CUSHION SYSTEM

UH-1B

The restraint system for this seat consists of a shoulder harness that converges into a single, strap-type inertia reel, and a lap belt, both with metallic end fittings that fasten into a quick-release buckle that is permanently fastened to a crotch strap. The crotch strap, in addition to preventing the occupant from submarining under the seat belt, retains the quick-release buckle in the seat pan.

The inertia reel conforms to Military Specification MIL-R-8236B(ASG) and is Type MA-6 under this specification. The strap is made of Dacron webbing per Military Specification MIL-W-25361(USAF), to which it must conform as stated in MIL-R-8236B(ASG), the inertia reel specification. Measurements of the strap gave a width of 1.75 inches and 0.082 inch in thickness; therefore, it appears to be Type III of MIL-W-25361(USAF). Type III webbing is required to have a breaking strength of 7000 pounds; except for the crotch strap, the remainder of the restraint system seems to be made of nylon webbing. The shoulder harness was measured to be 1.75 inches in width and 0.095 inch thick; the lap belt, 1.75 inches by 0.120 inch thick; and the crotch strap, 2.0 inches by 0.066 inch thick. The crotch strap appears to be made of Dacron. Specific data on restraint system materials are not available.

Both seat bottom and back cushions are made of urethane flexible foam having densities of 2 pounds per cubic foot for the back cushion and 4 pounds per cubic foot for the seat bottom cushion. The seat bottom cushion is contoured to fit the occupant's posterior, with a thickness of about 3.25 inches in the area of the thigh position, tapering to about 1.75 inches in the area of maximum body support of buttocks area.

Universal

This restraint system, except for the inverted "vee" belt, is composed of off-the-shelf military belts and metallic hardware. It consists of a shoulder harness that converges into a single strap inertia reel, a chest belt, and an inverted "vee" belt.

The lap belt is 3 inches wide and 0.087 inch in thickness and is made of nylon webbing conforming to Military Specification MIL-W-8630, Type III, which requires a minimum breaking strength of 8200 pounds. Two thicknesses of this webbing are used to form the belt to reduce elongation under load.

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Dacron webbing, 1.687 inches wide and approximately 0.085 inch tnick, is used for the inverted "vee" belt. It conforms to Military Specification MIL-W-19078, which requires a minimum breaking strength of 6000 pounds.

The inertia reel is a strap type and conforms to Military Specification MIL-R-8236B(ASG), Type MA-6. The strap is of Dacron, is 1.75 inches wide by 0.082 inch thick, and conforms to Military Specification MIL-W-25361 (USAF), Type III, which has a minimum breaking strength of 7000 pounds. A standard belt assembly, AN6506, modified to fit the dimensions of the seat is used for the chest strap. This belt uses nylon webbing, 1.75 inches wide and 0.090 inch thick, conforming to Military Specification MIL-W-4088, Type VII, which has a minimum breaking strength of 5500 pounds. The shoulder harness and the "vee" belt have metal loops that tie into the lap belt buckle. The chest belt has its own buckle; thus, ingress and egress require manipulation of two buckles--if the chest belt is used.

The seat cushions were made of polyurethane foam. The seat bottom cushion is 3 inches thick and is approximately 2 pounds per cubic foot in specific weight. These physical characteristics are sufficient in magnitude to cause the dynamic magnification of the seat occupant acceleration relative to the seat vertical acceleration.

CH-34

The system consists of two inertia reels of the aircraft cable type which attach to the shoulder harness, a seat belt, and an inverted "vee" belt. The metallic end fittings of the shoulder straps, the "vee" belt, and one side of the lap belt are placed one on top of the other, over a specially designed post-plate fitting that is attached to the opposite side of the lap belt. This restraint system was designed to withstand somewhat greater loads than the other designs evaluated. No failures were experienced in tests, and the system elongation was significantly less than for the other seats.

Two cable-type inertia reels are used. They conform to Military Specification MIL-R-8236B(ASG), Type MA-2, and are connected to the ends of the walking beam. The shoulder straps converge to a common fitting which, in turn, is fastened to the center of this beam. The shoulder harness and inverted "vee" belt are nylon webbing per MIL-W-8630-C, Type VI, and are 1.75 inches wide by 0.110 inch thick. This webbing is required to have a minimum breaking strength of 8700 pounds. Two thicknesses of nylon webbing are used for the lap belt per MIL-W-8630-C, Type II, sewn together. Singly, this webbing is 0.080 inch thick and 3 inches wide and has a minimum breaking strength of 8200 pounds.

Both seat back and seat bottom cushions are tapered. The seat back cushion is made of polyurethane foam with a thickness of 1.5 inches at the top of the seat back and tapering linearly to 3 inches at the bottom end. The seat bottom cushion has a top layer of polyurethane foam of 1.25 inches constant thickness and a lower layer of expanded polystyrene that has a linear taper so as to make the seat cushion total thickness 3 inches at the front and

2.25 inches at the rear.

CH-47

This restraint system consists of a shoulder harness that fastens into the chest protector, a lap belt and a shoulder harness using a single inertia reel. The system differs from the others in that the chest protector is a part of the restraint system.

The strap type inertia reel is identical to the one used for the Universal Seat. Its location, at the bottom of the seat back, is unfavorable in that it introduces too much total stretch into the restraint systems.

The shoulder harness and straps from the adjusters to the chest protector latch are made of 1.75 inch-wide nylon and conform to Military Specification MIL-W-4088E, Type VIII. A CH-47 drawing requirement states that the harness shall be capable of withstanding a dynamic load of 4000 pounds for 0.1 second duration without failure. The ends of the straps from the shoulders downward across the body end in metallic loop fittings that fasten into a single lever release fitting at the lower end of the torso shield.

The chest protector has a metallic fitting that fastens into a mating metal fitting at the front edge of the seat pan. The chest protector fitting is a 0.5-inch-diameter rod over a lower portion of its length, and this is directly exposed to the occupant. This fitting is the only object that would prevent the seat occupant from submarining underneath the seat belt. Thus, in a crash situation involving submarining, the fitting would cause extreme discomfort to the posterior portion of the human anatomy. The lap belt is a Military Standard, MS22033.

The seat bottom and back cushions are made in one continuous piece of resilient polyurethane foam, 1 inch in thickness. Because of the continual failure of the restraint system in the tests, no assessment of the performance of this cushion is possible.

DYNAMIC TESTS

Dynamic tests of the four seat designs were conducted by the Aviation Safety Engineering and Research Division of Flight Safety Foundation, Inc. These tests were performed on a sled track located at the Deer Valley Airport, Phoenix, Arizona. Accelerations were provided by a rocket-steam propelled sled. The sled was designed with shoes on the fore and aft center line to ride the single rail track, with special outrigger shoes on each side. These outrigger shoes rode rails on each side of the center rail for the power stroke distance, the first 14 feet, and thereafter ran over the ground.

The seats were mounted on the sled in the proper orientation to yield the design vector components of acceleration. Anthropomorphic dummies, equipped with accelerometers to measure the accelerations in the vertical, longitudinal, and lateral seat directions, were placed in the seats and secured by

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the seat restraint system. Accelerometers were also placed on the sled to measure its accelerations in the same three perpendicular directions. Outputs of these accelerometers were fed into oscillograph recording equipment and recorded.

A complete description of the test facility and apparatus, and detailed data of the results of all tests, can be found in Reference 22. Only a summary of the tests and pertinent data, observations, and conclusions relative to an evaluation of the designs are included in this report.

The four seats were designed for approximately the same, but not identical, criteria. Variations existed in magnitude of load factors and also in the manner in which load factors were combined. The series of tests were essentially the same for each of the seats. Table II summarizes data for the test conditions.

There was considerable variation in the sled accelerations between tests. The accelerations for Test Condition III were slightly low and also were not sustained for 0.1 second. However, when all other events and factors are considered, it is concluded that the sled performance was satisfactory for this series of tests. In general, the test data appear to be valid.

TABLE II

DYNAMIC TEST CONDITIONS

Test Condition	Loading	Deceleration Levels (g)	Time Duration
I	Simultaneous Vertical and Longitudinal Deceleration	Vertical g_V -12.5 Longitudinal g_L -22.5	0.1 Sec.
II	Simultaneous Lateral and Longitudinal Deceleration	Lateral g_{La} -22.5 Longitudinal g_L -22.5	0.1 Sec.
III	Simultaneous Vertical and Longitudinal Deceleration	Vertical g_V -25 Longitudinal g_L -45	0.1 Sec.
IV	Simultaneous Lateral and Longitudinal Deceleration[1]	Lateral g_{La} -45 Longitudinal g_L -45	0.1 Sec.

(1) The UH-1B, Universal, and CH-47 seats were not subjected to this test condition, as they were damaged beyond economical repair during Test Condition III.

The primary objective of the tests was to determine whether the seats would withstand the test loads and limit the accelerations on the occupant to

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values within human tolerance. In order to accomplish this, it was necessary to maintain the structural integrity of the seat for the design load conditions. None of the four seat designs tested fulfilled all of the design objectives. The energy absorption devices functioned in the intended manner in only one of the seats in one test. In most of the tests, the measured accelerations in the dummy exceeded presently accepted human tolerances and the design objectives. Dynamic magnifications, induced by the restraint-cushion systems used, contributed significantly to the high accelerations in the dummy. Excessive elongation in the systems allowed the dummy to assume a position well forward for the combined load conditions. The loads on the seat and therefore the loads on the energy absorbing devices were not as predicted. This, however, does not indicate that they would not have performed satisfactorily if they had been loaded differently. Other problems, such as load link failures, structural failure, restraint system failures, and disengagement of one seat from the track, negated the value of data for some of the tests.

The results of these tests are not considered to be necessarily representative of the crash protective capabilities of the seats. It would be most unfair to conclude that the design would not perform as intended. Each of the concepts has some desirable features. The most significant conclusion resulting from these tests is that they vividly displayed the need for a better occupant restraint-cushion system. The important design criterion for a desirable system is that it should be designed primarily for minimum deflection and not on the basis of strength alone. The restraint harness for the CH-34 seat was designed for much greater loads than the others and, therefore, displayed less elongation until the dummy accelerations exceeded human tolerance levels.

A comprehensive study was made of the tests and the test results. Following are brief comments on each of the seat designs and the test results. The acceleration data referred to in these comments can be found in Reference 22. Conclusions are not significantly different from those presented in Reference 22. Test Condition II is not discussed. This test was a combination of longitudinal and lateral accelerations. The tests were plagued with instrument malfunctions, load link failures, and one harness failure. Very little can be concluded from this test. It was not a condition in which the energy absorbing devices should have operated because of the absence of vertical load.

UH-1B SEAT

The maximum dummy accelerations for Test Condition I occurred at 0.05 second, while the maximum sled and seat accelerations occurred at 0.03 second. This agrees with the photographs, which show the dummy "sitting in space" until the restraining harnesses become taut. The faired value is about 23g for the dummy vertical acceleration, so that the magnification factor is 2.3 (23÷10). The indicated low (25g, maximum faired value) longitudinal dummy response is attributed to "an apparent instrumentation malfunction" (Reference 22, page 35). However, there is no more reason to suspect the validity of this record than there is any of the other records on the dummy.

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In Test Condition III, the UH-1B seat experienced a premature failure at approximately 0.04 second. After a careful study of the sequence of photographs and an inspection of the failed seat, it was concluded that the failure was due to faulty hardware. A nut became disengaged from a bolt attachment at the apex of the triangular support. The failure of the nut precipitated subsequent failures that led to the destruction of the seat. The nut was recovered from inside the tubular structure and subjected to hardness tests. There were indications that this nut had strength equivalent to only approximately 50 percent of the specification value. Checks of other nuts of the same series used on the seat indicated sufficient strength.

Because of the loss of the structural integrity of the seat, the accelerometer data are not considered to be exemplary of the seat as an entity, but more likely representative of the failure structure.

UNIVERSAL SEAT

The accelerometer records for Test Condition I are very oscillatory in nature, except for the sled, with the seat accelerometer tracing indicating nonlinear vibration. This rather erratic response may be caused by the behavior of the dummy. The dummy quickly assumed a position on the forward part of the seat pan, and later a part of the shoulder restraint system failed.

The apparent amplification factor for the dummy in the vertical direction was 2.9 (32÷11), while the longitudinal was 1.1 (31÷28), all computed from faired values of the traces. According to the test report, Reference 22, page 52, the seat belt load link on the right side failed at 0.09 second.

In Test Condition III, the seat started moving upward and forward parallel to the floor mounting rails almost immediately after firing.

The photographs and the seat accelerations indicate that the seat was completely airborne at 0.05 second and received no further acceleration from the sled. As the seat moved upward and forward along the floor mounting rail, the energy absorber functioned. This was the only energy absorber that functioned in this series of tests. This failure pointed out rather conclusively the need for a positive locking device for the seat to the track. The sled velocity change at this time was 1.76g-sec, which yields about 35g average acceleration. At the 28-degree orientation of the seat, the horizontal component is 16.4g. The peak value during this same time period was approximately 56g at the apex of the wave form. This yields a vertical component of 26.3g on the seat. In view of these facts, there is not much reason to assume that the seat experienced anything over 20g for any longer than instantaneously.

CH-34 SEAT

The vertical as well as the longitudinal accelerations of the dummy for Test Condition I seemed incredible. These records indicate magnification by a factor greater than two. The motion pictures and still photographs do not

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seem to corroborate the situation just described. They show that the restraint system held the dummy very well during this test and that the system is not nearly as softly sprung as the accelerometer records indicate.

There is no doubt that this restraint system was best in performance, yet the accelerometer records for the dummy show large peak values. These traces for longitudinal and vertical response of the dummy indicate non-linearity. The vertical amplification factor based on the peak value for the dummy is 6 (60±10), while it is 3.5 (35±10) for the peak faired value. The "spiked", a large amplitude, initial pulse resembles a parabolic cusp except that it is not pointed, but has a rounded top. It can be approximated with a triangle. The RMS value for a triangle is $1/\sqrt{3}$ times the peak values, and, applied to this particular case, yields an amplification factor of 3.46 (60±10$\sqrt{3}$), which is close to the faired value. The longitudinal amplification factor for the dummy is 3.8 (95±25) for the maximum peak faired value. Again, the large amplitude, initial spike lends itself to approximation by a triangle and the amplification factor is 2.2 (95±25$\sqrt{3}$) for this RMS value; this is to be compared to the faired value of 2.08.

The RMS value of the dummy vertical acceleration as used above is 34.6g considered to be applied over a 0.05-second time interval, and the longitudinal value is 55g RMS at an effective time of 0.02 second. Both of these are "borderline" and probably exceed the limits of human tolerance. Location of these points on Figure 17, page 74, and Figure 2, page 58, respectively, of Reference 11 places them in the area labeled "area of moderate injury".

In Test Condition III, the CH-34 seat again exhibited high peak accelerations on the dummy. There were no structural failures. Loads great enough to shear the safety pins and actuate the energy absorbing strut were not attained. Friction between the movable track and the fixed support was neglected in the design calculation of the force necessary to shear the pins and thus to actuate the energy strut. This friction can be quite high, and undoubtedly its omission in the design analysis accounted for the failure of the energy absorber to function in the dynamic tests. The track components are fabricated of 7075-T6 aluminum alloy, and the sliding surfaces did not receive any friction reducing treatment. They were coated with zinc chromate, and the resulting friction coefficient was very high.

CH-47 SEAT

As noted in Table I, this seat was not designed for combined vertical and longitudinal accelerations. However, it received the same test accelerations as the other seat designs, which does seem more realistic. As in all three test conditions, failure of the chest protector and other restraint components caused the dummy to leave the seat immediately after the sled began to move. In this design, the chest protector is a part of the restraint system. Also, the test report, Reference 22, page 91, states that the seat-belt-to-seat-pan attachment on the right side failed.

The photographs show that the dummy left the seat at 0.093 second and that

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the shoulder harness failed at approximately 0.106 second. Accelerations on the dummy will not be discussed because they are not significant on account of the restraint system failures.

The shoulder harness failed in Test Condition II, and the dummy jackknifed over the lap belt. Failure of the shoulder harness was at a point where the web strap p_. ed over the rollers in the harness adjust fitting.

Restraint system failure again occurred in Test Condition III. The shoulder harness failed, and the seat belt fittings pulled out at the attachment to the seat. There was evidence of some energy absorber action. However, this is not significant. It was considered to have been caused by eccentric loads due to the dummy's leaving the seat.

DESIGN EVALUATION

The primary objective of the four seat designs was to provide an armored seat that would keep the occupant from experiencing accelerations, in the event of a crash that exceeded accepted human tolerance levels. Different approaches and concepts were used for each of the designs. All are results of unique ingenuity and considerable analysis and design effort in their development. The four seats were designed for approximately the same, but not identical, criteria. Variations existed in magnitude of load factors and also in the manner in which load factors were combined. Each set utilized different concepts of basic design and crash load mitigation devices. The series of tests were essentially the same for each of the seats.

None of the four seats tested fulfilled all of the design objectives. The energy absorption devices functioned in the intended manner on only one of the seats in one test. This does not indicate that they would not have worked had they been loaded differently. It is concluded that with the exception of the CH-34 seat, they would have worked in the intended manner if the restraint system had held the dummy in the proper position in the seat. Excessive elongation in the system allowed the dummy to assume a position well forward and did not load the seat and therefore the energy absorber in a manner for which it was designed.

It is believed that the energy absorbing device in the CH-34 seat would not have functioned under the design loads. The actual friction loads, as determined by analysis, are of such magnitude as to preclude satisfactory operation of the energy absorber. The manufacturer's design analysis did not include the effects of friction. The restraint system on this seat was designed for much higher loads than the others and therefore resulted in somewhat less elongation.

Although none of the designs fulfilled the design objectives, each had good design features, and much useful information was gained from the design effort and the dynamic tests. One of the most significant conclusions was that an occupant restraint-cushion system must be developed as a companion to the development of a crash load mitigation seat. Deficiencies in the restraint systems and their effect during the tests prevented a good

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evaluation of the crashworthy properties of the seats.

Following are comments on each of the designs. Table III is a weight comparison.

UH-1B

- The design concept utilizing tubing and sheet metal type construction is considered to be efficient, easy to fabricate and probably lowest in cost.

- The complete unit design concept allows easy adaptation to other aircraft.

- The energy absorbing device is a cable-sheave type and should demonstrate ideal load limit characteristics.

- The energy absorber is unidirectional.

- Sliding friction is present during seat movement. It was evaluated and accounted for in the design, but the magnitude of the friction used was too low.

- Means of vertical adjustment are not provided.

- Restraint system allows excessive elongation.

- The single-strap type inertia reel location on the support structure contributes to the excessive elongation in the shoulder harness.

- The single-lever, quick-release buckle on the restraint harness allows rapid postcrash egress.

- The cushion is considered to be too thick. It allows relative velocity between the occupant and the seat and results in excessive accelerations on the occupant.

- The design and placement of armor allows the occupant more freedom of movement for control of the aircraft and more comfort while doing so. Seat bottom and back armor was placed on the seat. Protective armor for other directions was to be on the aircraft structure.

- Seat bucket armor in the seat bottom consists of two pieces that can be easily replaced or removed by removal of two screws. In the seat back, there are nine separate pieces. Complete removal of this armor requires the removal of 24 screws. While the number of screws to be removed for complete removal of the armor is a disadvantage, the separate pieces offer the advantage of separate replacement of only those that

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are damaged.

UNIVERSAL SEAT

- The stiffened sheet metal, integral armor is relatively simple and easy to fabricate.

- The complete unit design concept allows easy adaptation to other aircraft.

- The energy absorbing device is a VanZelm metal bending type.

- The energy absorbing device is bidirectional.

- Friction is adequately accounted for and reduced by the use of rollers for fore and aft loads. Sliding friction is present for side loads.

- The restraint system allows excessive elongation.

- Fore and aft and vertical seat adjustment is provided; vertical adjustment is at the expense of reducing the energy absorption stroke.

- The placement of the armor affords good protection and allows relative freedom of movement.

- A ballistic hit in the seat bucket armor requires the removal and replacement of many screws for replacement of the damaged piece, since the armor is integrated as structural material.

CH-34

- The seat is designed specifically for the CH-34 aircraft. It depends on a bulkhead or the equivalent for support. It is not readily adaptable to other aircraft.

- The structure is simple, rugged and efficient.

- The energy absorbing device utilizes crushing of honeycomb material to be loaded after a shear pin fails at a predetermined load.

- The energy absorber is unidirectional.

- Sliding friction is present between two zinc chromate primed aluminum surfaces. The magnitude of frictional loads is such as to preclude operation of the energy absorber, as intended. Friction was not accounted for in the analysis.

- The restraint system possesses high strength and relatively low elongation.

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- The buckle termination of the harness requires the build-up of four thicknesses of steel harness terminating on a single pin and then the insertion of a locking pin. This would tend to prevent rapid egress.

- One piece of armor held in place by four hexagon head bolts is used in both the seat bottom and back. While this makes the armor easily removable, it requires replacement of the complete panels in case of ballistic damage.

CH-47

- The design concept utilizes molded ceramic armor materials for protection and structure. This seems to offer no outstanding advantage except good ballistic protection by placing the armor in close proximity to the seat occupant.

- An inertia reel and strap from the top of the seat back to aircraft structure are used to restrain the seat. This is considered to be objectionable in that it requires a bulkhead or some other structure behind the seat for strap attachment.

- Energy absorption is accomplished by crushing three rows of blocks of Trussgrid material.

- Energy absorption capability is unidirectional.

- The seat and cushion are comfortable.

TABLE III
WEIGHT COMPARISON OF SEAT DESIGNS (POUNDS)

| Seat Design | Structure | Armor | Movable Part | | Support | Complete Seat |
			Restraint Cushion System & Misc.	Total Movable Part		
UH-1B	30.6	71.8	4.9 [1]	107.3	35.0	142.3 197.1 [2]
Universal	74.7	44.1 [3]	(4)	118.8	42.0	160.8
CH-34	43.9	107.0	18.3	169.2	30.3 [5]	200.0 [6]
CH-47	16.5 [7]	109.1	15.5	140.6	18.1	158.7

(1) Cushion only - restraint system included in structure.
(2) Includes 50 percent of 109.5 pounds of armor protection located on the aircraft.
(3) Removable armor only. Includes shoulder, front torso and side armor. Seat bottom and back armor used as structure and is included in weight of structure.
(4) Included in structure.
(5) Seat designed for installation in CH-34 aircraft. 30.8 pounds represent 50 percent of the modification weight necessary for installation.
(6) Includes modification weight noted in (5).
(7) Armor was extensively used as structure - other structural weight is negligible.

DESIGN CRITERIA FOR IMPROVED CREW SEAT

The primary objective of a crashworthy seat design is to keep the occupant from experiencing accelerations, in the event of a crash, that exceed human tolerance levels. This must be accomplished by restraining the occupant in a seat employing crash mitigation properties and possessing sufficient strength in the tie-down chain to restrain the occupants up to a point at which survivable impact conditions no longer exist.

Originally, the specified design load criteria required load attenuation in the vertical direction only to limit the loads on the occupant to 20±5g in the vertical direction. The seat was also required to have sufficient strength in the longitudinal and lateral directions to withstand a static ultimate load factor of ±45g. A preliminary design of a seat meeting these basic requirements was made. It was quite heavy and was not considered to be the optimum seat for crash survival because of its having load attenuation capabilities in only the vertical direction.

A modification to the contract revised the design requirements to include load attenuation capabilities in the vertical, longitudinal, and lateral directions and significantly reduced the design load criteria. These new design requirements are as outlined in a report by Aviation Safety Engineering and Research, Reference 19, entitled "Proposed Military Standard for Improved Crew/Passenger Survival Seats and Body Retention Systems". It is the result of a program of research and investigation by AvSER under contract with the U.S. Army.

The requirements of Reference 19, as revised by discussions with USAAVLABS and AvSER personnel, were made applicable to this development program by contract modification. These and other requirements are summarized in this section. The energy absorbing system is to be tailored such that it will function, so far as practical, throughout an occupant weight range from 5th percentile to 95th percentile man, with and without armor, without exceeding the deceleration limits. The seat design shall include provisions for airframe installation in the UH-1B aircraft.

LONGITUDINAL STRENGTH AND DEFORMATION CHARACTERISTICS

The initial seat collapse strength under longitudinal loading shall exceed 22g, but shall not exceed 35g. In addition to meeting the requirements for initial collapse load, seats shall be designed to maintain load after initial collapse occurs. The seat shall be so designed and constructed that its load deflection curve carries into the shaded region above curve B of Figure 9 without first falling below the base curve, A.

VERTICAL STRENGTH AND DEFORMATION CHARACTERISTICS

The seat shall exhibit vertical static strength such that initial vertical collapse occurs under vertical loading of 17±2g based on an effective occupant weight of 160 pounds (80 percent of total weight of 200-pound occupant). The seat must be capable of vertical deformation of 6 inches while maintain-

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ing a vertical load of 17±2g. A vertical deformation of 12 inches was required by the basic contract. AvSER recommends that 6 inches be a minimum, and more is highly desirable.

LATERAL STRENGTH AND DEFORMATION CHARACTERISTICS

The seat shall exhibit lateral static strength such that initial lateral collapse occurs under lateral loading of 4000 pounds and shall have a lateral deformation of not less than 2 inches while maintaining a load of 4000 pounds. Tolerance on the lateral load is ±400 pounds.

PROOF TESTING OF SEAT STRUCTURES

The seat will be static tested under longitudinal, vertical, and lateral loading. These loads will be applied separately at the location of the center of gravity of the occupant. Requirements for load deflection behavior of the seat need only be fulfilled for a fully loaded seat. Deflection will be measured at the point of load application.

The seat will also be tested under a simultaneous loading consisting of loads of the following ratios: vertical, 100 percent; longitudinal, 75 percent; and lateral, 50 percent.

PROOF TESTING OF SEAT ATTACHMENTS

Seat-to-structure attachments will not fail under loading conditions equivalent to simultaneous application of maximum forward and lateral loads.

OCCUPANT RESTRAINT SYSTEM

The restraint system for rotary-wing forward-facing crew and passenger seats shall include a lap belt, a dual strap shoulder harness, and a lap belt tiedown strap. The restraint system will, in all cases, provide alignment and support of the occupant throughout a survivable crash situation. Loss of support will not occur due to stroking of seat energy absorbers and/or plastic deformation of the seat. The minimum design loads and dimensional requirements for all restraint system components are as listed in Table IV.

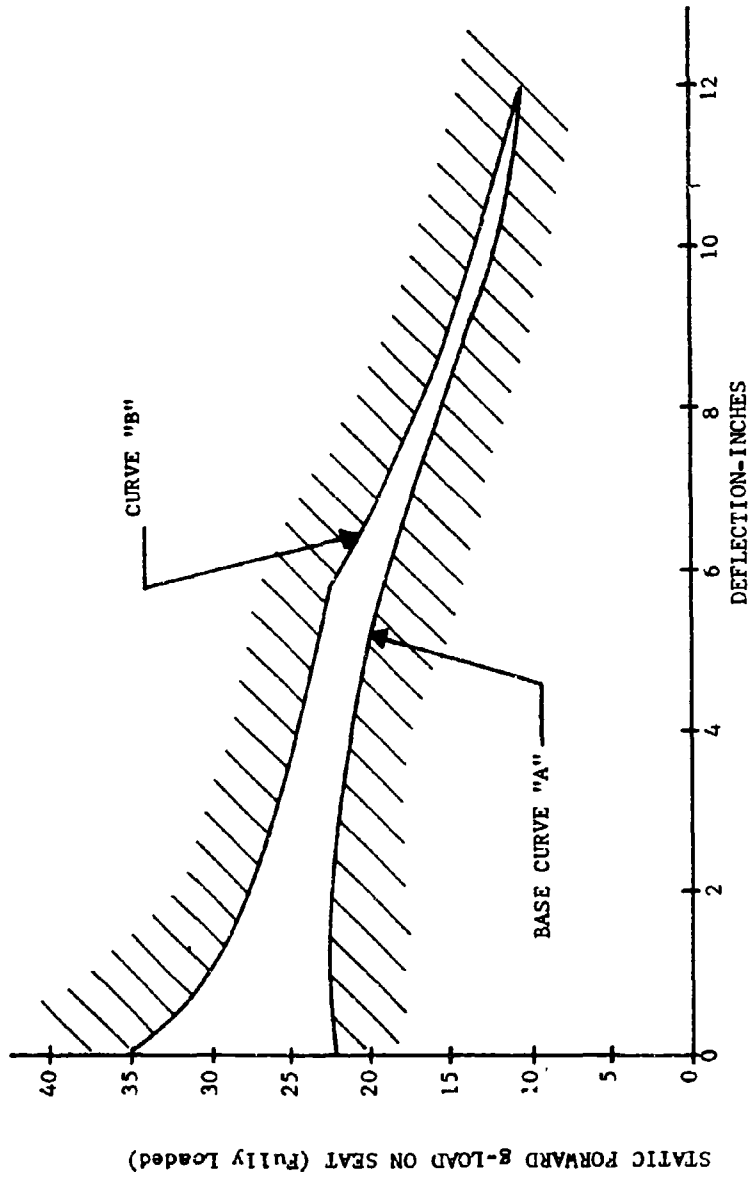Figure 9. Envelope of Load Deflection Requirements for Forward-Facing Crew and Passenger Seats.

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TABLE IV

RESTRAINT SYSTEM DESIGN LOADS AND DIMENSIONS

	Minimum Ultimate Design Load[1] (lb.)	Minimum Width (in.)	Minimum Thickness (in.)
Lap Belt Strap	5000[2]	3.0	0.15
Shoulder Harness Strap	3600[3]	2.0	0.10
Lap Belt Tie-Down Strap	2200	1.75	0.10

(1) Maximum elongation of all strap components including terminal hardware shall not exceed 10 percent of the original length of the strap when loaded to these design loads for a period of 2 seconds.

(2) Loop strength.

(3) Total harness load.

The occupant restraint system must also meet the following requirements:

● Quick release hardware allowing a one-point release of all restraint will be provided.

● Hardware components in all restraint release mechanisms will carry the design loads listed in Table IV without permanent deformation.

● Belts, harnesses, or straps shall be attached to the seat and/or basic aircraft structure in such a manner as to preclude premature failure due to stress concentrations because of misalignment of components during any possible seat deflection and/or body orientation during the crash sequence.

● The lap belt shall pass over the pelvic region of the occupant at an angle of 45° to 55° measured from the plane of the surface of the seat pan.

● The shoulder harness shall pass over the shoulders at an angle of 0° to 30° above a perpendicular to the spine.

● Load distributing pads shall be used to separate hardware components from direct contact with the occupant in locations where acceleration loads will lead to high contract pressures.

● No castings will be used in load-carrying elements of the restraint system. All hardware shall be fabricated from materials having an ultimate elongation of 10 percent or more.

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● Inertia reels shall be capable of withstanding the full shoulder harness load of 3600 pounds.

SEAT CUSHIONS

Seat cushions will be provided for all seats in which the absence of such cushions would allow contact of the occupant with injurious components during deformation of the seat. Seat cushions should provide maximum comfort compatible with occupant protection requirements. Crew seats will be form fitting to insure maximum comfort. A maximum of 1.5 inches of flexible foam shall be used in the comfort pad or top layer of the cushion. The lower portions of the seat cushion will be fabricated from permanently deforming energy absorbing material. This material shall crush under a total load of 2000 to 3000 pounds.

SEAT ATTACHMENTS

Attachment of the seat to the basic aircraft structure shall have sufficient strength to preclude failure under the maximum loads as specified above. In addition, attachments shall be made in such a manner that crew members are provided with obvious warning whenever seats are not positively locked in place.

SEAT ADJUSTMENT

The seat shall have an adjustment of 5 inches vertically and 10 inches horizontally.

BALLISTIC PROTECTION

The seat shall be armored to provide 100-percent protection of the trunk-torso portion of a 95th percentile occupant against 7.62mm APM-61 ammunition fired at 100 yards range impacting at 15° obliquity. The armor shall not be integrated as structure.

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DESIGN STUDY

The design criteria discussed in the previous section combine more necessary and desirable features for an armored crashworthy aircrew seat than have been used for any known seat design effort. Some of the requirements are opposing. The wide range of weight configurations from a 5th percentile man without armor to a 95th percentile man with armor under which the energy absorbing devices must function to limit the loads on the occupant to varying values in three different directions makes the problem quite complex. The design problem is further complicated by the required and necessary height and the fore and aft position adjustment capabilities. The ever-present general problem of obtaining the necessary clearances relative to other items in the aircraft influenced the movement for energy absorption and the placement of the armor. In addition to the specific requirements as stated, it was necessary to design for minimum weight and ease of fabrication at low weight. The final design presented herein will meet all of these requirements.

The primary objective of this design effort was to develop a seat that would limit the loads on the occupant in the event of a crash to values within human tolerances. Most of the design effort was directed at this problem. Many compromises had to be made in establishing the basic concept. Many concepts investigated yielded excellent results and capabilities for loads in one direction, but were wholly inadequate for loads in the other direction because of interaction between the loads and energy absorbing systems.

Therefore, two basic and interrelated problem areas were paramount. What basic energy absorbing concept should be used and what type of energy absorbing device would best fulfill the requirements? The improved design could constitute a new design or be a compilation of one or more of the previous designs that were evaluated. Each of the four designs had some good features, but none would meet the specified requirements. The UH-1B, Universal, and CH-34 seats used the same basic concept of utilizing a support for the seat bucket. The bucket moved relative to the support for energy absorption stroke. The CH-47 design was different in that it utilized the energy absorbing device for the support. A comprehensive discussion of the tests is included in the section entitled "Dynamic Tests".

Many concepts to accomplish the objectives were considered and investigated to varying degrees. The concept that most conceptual designers think of first consists of a basic structure that will sustain loads up to a specified magnitude and then deform in a controlled manner over a long distance to limit the loads on the occupant. This type of structure is idealistic but was investigated extensively. From the initial conceptual studies, it was concluded that a concept utilizing the deformation of the basic structure for energy absorption was impractical for a seat that must offer crash protection for a wide variation of weights for the required range of seat-occupant-armor configurations. The concept could undoubtedly be developed for a specific load input for one occupant and weight configuration. However, this would not fulfill the requirements for a practical seat for use

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in an aircraft wherein more variables and special features must be considered.

After much preliminary study, analysis, and evaluation of design concepts relative to requirements, it was concluded that the optimum seat should be based on the following:

- The seat to be supported from the floor of the aircraft.

- Energy absorption devices to be located between the seat bucket and the support frame for vertical and lateral stroke and between the support and floor for forward stroke.

- This support structure to have sufficient strength to withstand all design loads without failure or deformation that would adversely affect the stroke travel or load capability of the energy absorbing mechanisms.

- Assume the occupant, seat bucket, and armor to act as a single mass for energy absorption analysis. Therefore, the restraint-cushion system must maintain the occupant in a fixed position under load.

The final design is based on these basic items. In this design, a framework of steel tubes supports the seat bucket. Tubular construction is considered to be the most efficient for this application. The tubes can be used for tracks on which the bucket can slide for the energy absorbing stroke. The bucket is allowed to move relative to the support in the vertical and lateral directions. The complete seat including the support moves forward relative to the aircraft floor for the forward energy absorbing stroke. A variation of the concept was investigated in which the seat was allowed to rotate forward for the energy absorbing stroke in the forward direction. This appeared feasible because the load limiting requirements in this direction range from 22g to 35g, and it was believed that a single energy absorber in the aft support tubes would limit forward loads within the specified range for the various weight configurations. Problems were encountered with the effects of stability and deflection of the support system for lateral and vertical movement when the forward energy absorber was actuated.

The parallel slides on which the seat bucket must move in the lateral and vertical directions would rack and negate movement in these directions. Designs that would eliminate this problem added more weight to the seat than the present design, in which the entire support remains intact.

DESCRIPTION OF IMPROVED AIRCREW SEAT

The design consists of a tubular steel structure which supports a seat bucket fabricated of aluminum alloy sheet. The support is made up of two triangular frames interconnected by cross members at each apex and by diagonal braces in the plane of the aft members. It was necessary to place the

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entire support structure aft of the seat bucket to allow sufficient clearance for lateral movement in the armored configuration and for vertical movement for the energy absorbing stroke. The seat bucket is supported on two cross members between the forward vertical tubes of the support. Fittings at the end of the cross tubes allow the seat to slide vertically on the vertical support tubes for the energy absorbing stroke. The seat slides on the cross tubes for the lateral energy absorbing stroke. Vertical adjustment of 5 inches is accomplished by a linear electrical actuator with the bucket moving on tracks attached to the support cross tube.

The seat bucket is designed to react the maximum loads from the harness, as shown in Table IV, and to transmit these loads to the support structure. The restraint harness consists of a lap belt, a crotch strap, and a shoulder harness, all terminating at a single point where the operation of a single lever releases all straps simultaneously. In an effort to reduce the lengths of the shoulder straps and, therefore, the total elongation of the system, the inertia reel is mounted on the top of the seat back. The seat bottom and back are curved, and thin comfort cushions as provided.

The complete seat, including the support structure, is adjusted fore and aft by movement along two floor-mounted tracks the same as presently in the UH-1B aircraft. Adjustment is positioned and locked by pins on the seat being engaged in holes in each of the tracks. For energy absorbing stroke in the forward direction, the seat moves on rollers. A forward roller on either side engages the floor tracks. An aft roller on each side engages a separate track that is locked to the floor tracks by the positioning pins.

Energy absorption is accomplished by a Hayes-developed device by bending steel rods around rollers in a controlled manner. Stroke distances of 7 to 12 inches are provided in the vertical direction, depending on adjusted height of the seat bucket, 2 inches in either direction laterally and 5.6 inches forward.

Ceramic-fiber glass composite armor developed by Cincinnati Testing Laboratories (CTL), Space and Technology Division of the Studebaker Corporation, Cincinnati, Ohio, is used for ballistic protection. A shell fabricated of flat panels surrounds the seat bucket to provide protection for the torso of a 95th percentile occupant. The armor shell is attached at the same points used to attach the seat bucket to the support and moves with the bucket when adjusted or in energy absorber stroke. The armor is not used as structure. Therefore, the seat can be used either with or without armor.

Figure 10 is a sketch of the complete seat without armor. Other sections of this report discuss the energy absorbers, ballistic protection, and restraint-cushion system in more detail.

ENERGY ABSORPTION

The seat, as the occupant's supporting structure, and the underlying floor structure are the media through which forces are transmitted to the occupant. The dynamic response of these media during an impact determines the

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Inertia Reel

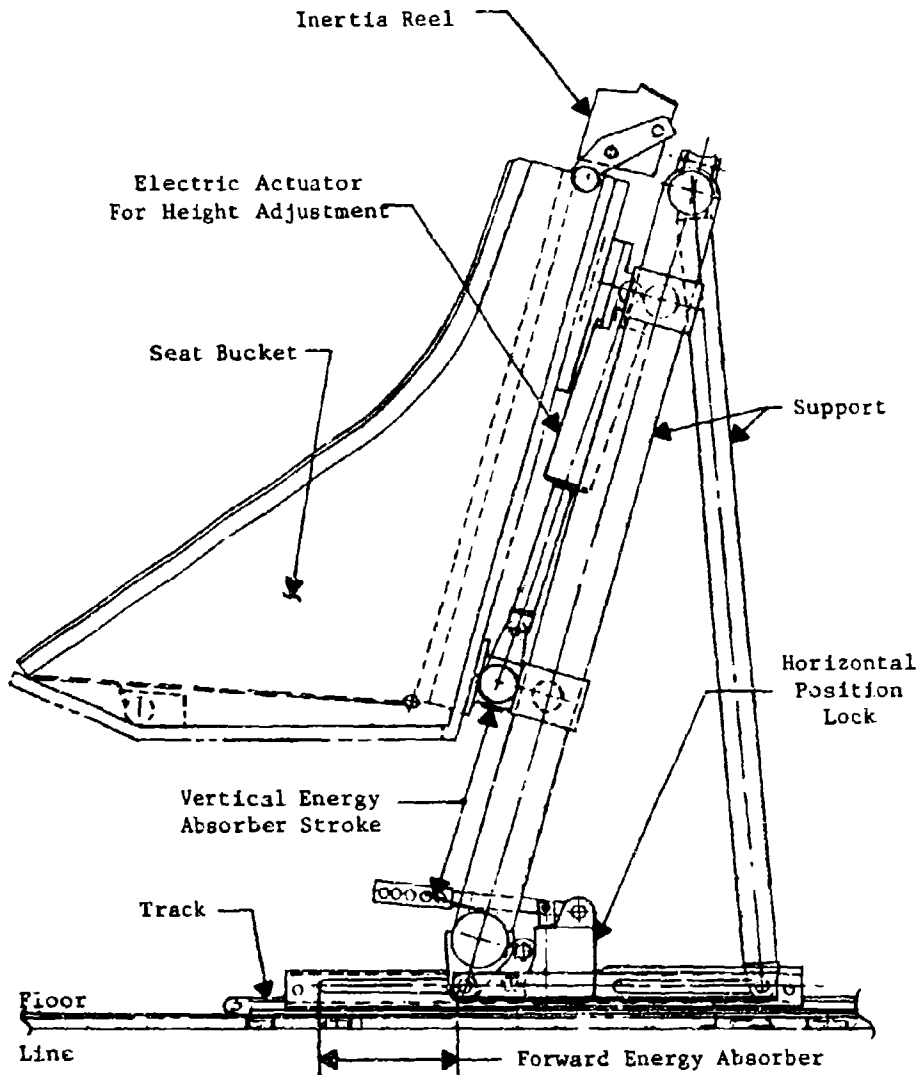Electric Actuator
For Height Adjustment

Seat Bucket

Support

Horizontal
Position
Lock

Vertical Energy
Absorber Stroke

Track

Floor

Line

Forward Energy Absorber

Figure 10. Hayes Aircrew Crash Survival Seat (Armor is
Omitted).

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Seat
Support
Assembly

Track
Enlarged For Clarity

Seat
Armor
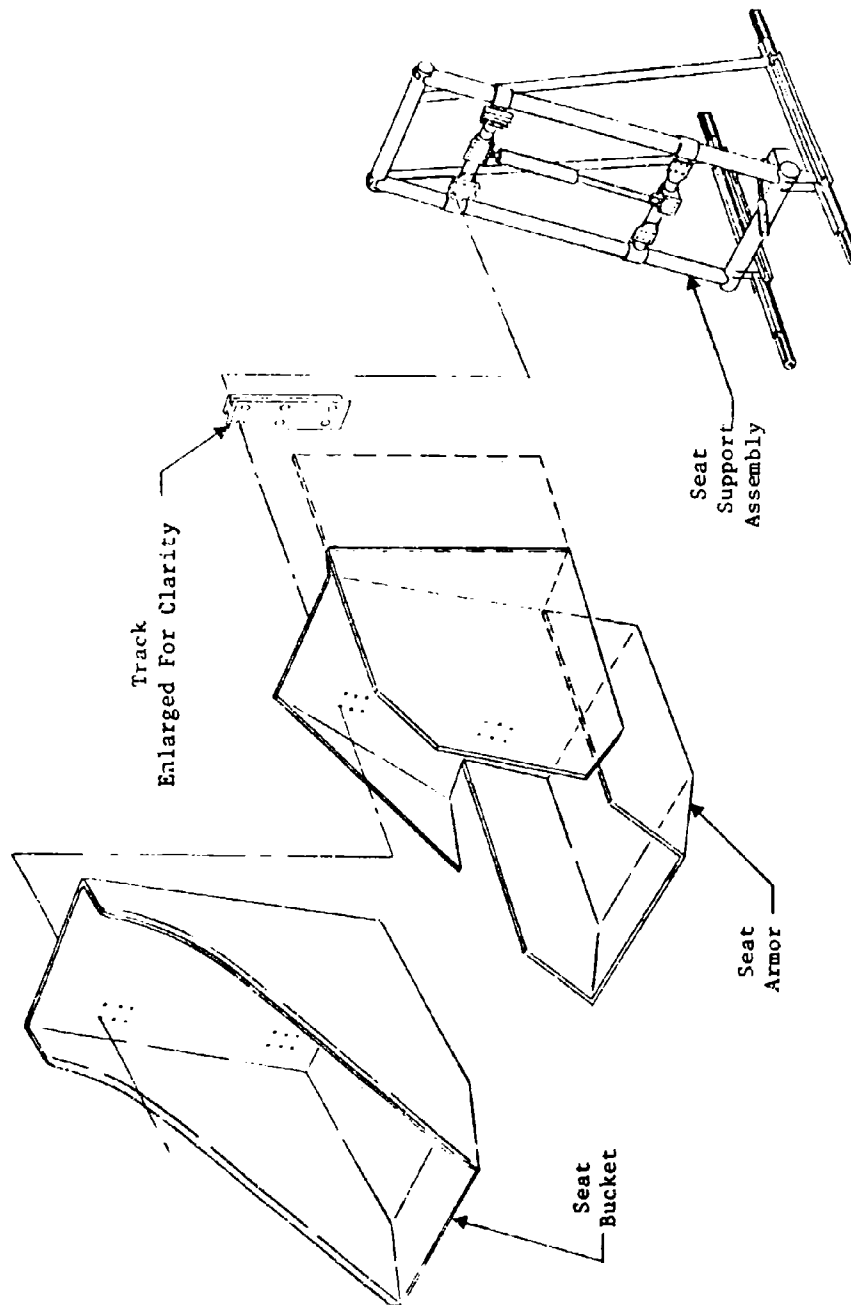
Seat
Bucket

Figure 10 (Cont.) Hayes Aircrew Crash Survival Seat (With Armor)

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manner in which the forces acting on the aircraft structure can be modified before reaching the occupant. An extremely rigid structure would transmit the forces without modification. An elastic structure, which has energy storing properties, may modify the magnitude and other characteristics of decelerative force to the extent that amplification takes place. A more desirable situation is that in which the structure has high energy absorbing characteristics. This ideal form of crash energy absorption results in attenuation of the crash forces transmitted to the occupant.

Of prime concern in impact decelerative loads is the possibility of receiving overshoot g-loading on various parts of the body. This may be caused by application of a force to the seat at a high onset rate which excites the resonant frequency of the seat cushion. Since the human body is a series of spring-mass systems, the seat cushion acts as an additional spring connection between the body and the seat. Excitation of the cushion spring resonant frequency causes an acceleration overshoot on part or all of the second spring-mass system - the occupant. This will result in what is referred to as a "jolt" load whereby "bottoming out" occurs before the impact velocity has been dissipated.

In designing an energy absorbing crash survival seat to attenuate decelerative forces, it is necessary to dissipate the kinetic energy at impact, $K.E. = \frac{1}{2}mV^2$. This can be accomplished best by controlling the seat stopping distance and time during which a design force will act. The efficiency of an energy absorber may be expressed as the ratio of energy dissipated to the initial energy to the system. Energy absorbed is equal to the area under the load-deflection curve, and it is readily recognized that a rectangular shaped load-deflection curve results in the least seat load and deflection in the case where all energy is dissipated. An elastic system, such as a mechanical spring, merely stores the kinetic energy, only to return it to the seat with the likelihood of force magnification. The spring load-deflection curves appears as a triangular shape, and it is necessary to increase seat load or deflection for storing the kinetic energy. A practical energy absorber will fall somewhere between the two curves representing the elastic and complete plastic systems. These principles are demonstrated in Figure 11.
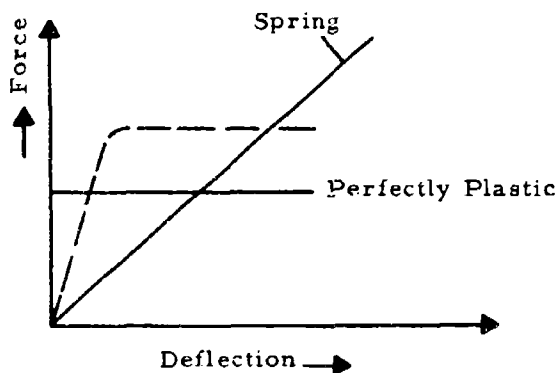

Figure 11. Typical Energy Absorbing Characteristics.

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The major variables of impact energy dissipation are velocity and stopping distance, and the quantities to be determined as far as man is concerned are maximum deceleration, duration, and onset rate of deceleration.

An efficient energy absorber must generate a constant force that is uniform throughout the entire stroke and independent of velocity. It should be capable of being installed for indefinite periods in any aircraft environment without having its energy absorption chracteristics affected and with no maintenance required.

The total energy that can be absorbed by an energy absorber at a constant load is a direct function of the length of the stroke available. A greater stroke distance will offer better protection for the occupant. In an aircraft, and especially the UH-1 helicopter, the space for available travel is obviously limited. Vertical stroke travel had to be limited to that above the floor line. Control mechanism and other systems under the UH-1 floor prevented the utilization of additional travel below the floor. Lateral travel is limited by the aircraft structure on the outboard side of both seats, by the cyclic control on the left side of both seats, and by the center control pedestal on the inboard side of the left seat. In the Hayes seat design, provisions are made for energy absorbing stroke travel of 5.6 inches forward, 2 inches in either direction laterally, and 7½ to 12½ inches vertically, depending on the height adjustment position. The outboard lateral movement is slightly reduced in the armored configuration.

Many types of energy absorbing devices were considered in the determination of the basic seat concept and for use in the specific concept as used in the final design. These included the devices used on the four seats that were evaluated as previously discussed. Others considered were hydraulic, pneumatic, springs, frangible tubing, expanding tubing, fabric occupant support, and various ways of metal bending. A rather unique device that absorbs energy by the cyclic bending of wire rings or torus rolling between two cylinders was investigated extensively.

It is a proprietary item of Aerospace Research Associates, Inc., (ARA), West Covina, California, with a patent pending.

Since this device is in tubular form and since tubes can be used on efficient seat support structure, it was investigated for use in a deforming structure concept as well as a separate energy absorber. The device is highly efficient but is fabricated of thin wall tubing. In all configurations using tubing efficiently, the tubes were loaded in bending as well as axially, and the thin tubing necessary for energy absorbing action did not have sufficient strength for the combined loading. The device can be used in the present seat configuration but is considerably more expensive than the energy absorber used.

These factors were evaluated:

- Energy absorption characteristics

- Efficiency

- Consistency of load vs deflection

- Size and space requirements

- Weight

- Maintenance requirements

- Adjustable load capability

- Availability

- Cost

- Simplicity

- Predictability

- Reliability

Configurations investigated utilizing crushing of material such as honeycomb and similar materials showed disadvantages in space limitations for energy absorbing stroke and seat adjustments. Hydraulic and pneumatic systems were investigated primarily because they are capable of being made adjustable for various loadings. They were eliminated for several reasons: they require considerable development work, they require periodic maintenance in service, their cost is relatively high, and they are heavier. Springs with locking devices to prevent the release of the stored energy were also eliminated because of weight and large size.

Considerable interest has been shown in fabric seats because of the comfort features they offer for the occupant. A seat bottom of fabric can be designed that will be an effective energy absorber. However, it would meet the vertical load requirements for an occupant in only a single weight category. Fabric seats were eliminated from consideration primarily because of the loosening of the occupant restraint when they are deflected for energy absorption.

Controlled metal bending appears to offer more advantages than other devices considered. A new device in which energy is absorbed by the controlled bending of wire was developed and tested by Hayes and is used in the seat design. It has several advantages: it is simple; load-deflection curve approaches the ideal; it is highly predictable; friction is greatly reduced; no maintenance is required; lightweight, energy absorbing elements can be changed to accommodate various weight configurations; and the cost is low.

Figure 12 is a sketch of the energy absorber used for the vertical direction. It consists of eight elements as shown and is adjustable for four different occupant-armor configurations in order to limit the vertical load
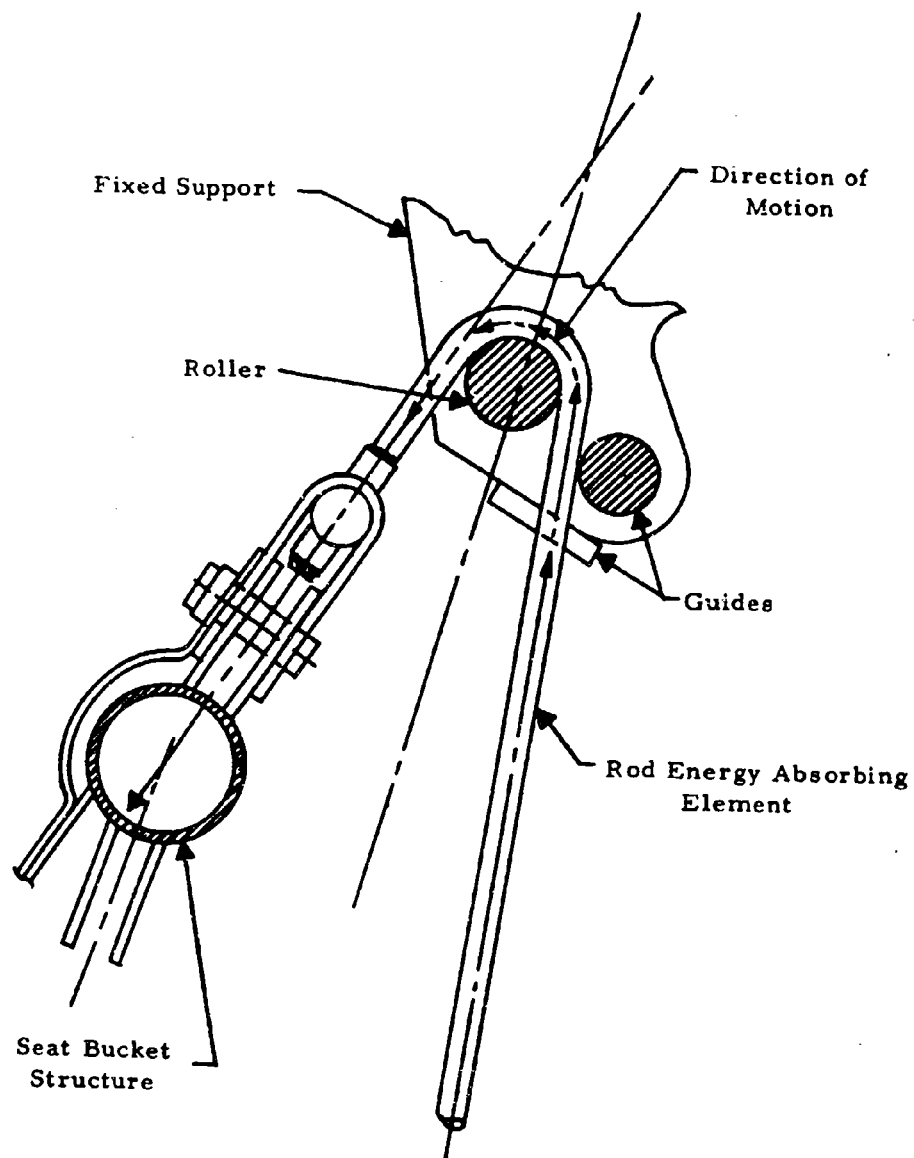
38

Figure 12. Bending Rod Energy Absorber.

on the occupant to 17g. The energy absorber in the lateral direction is
similar but is not adjustable. The rod energy absorbing elements in the
forward direction are also similar, except that they bend and rebend around
two rollers for additional load capability. They are not adjustable. The
elements can be replaced to use a different element for seat configurations
with and without armor.

Appendix II contains a detailed discussion and analysis of the energy ab-
sorbing device.

RESTRAINT-CUSHION SYSTEM

Maximum protection against crashes and other abrupt decelerations can be
achieved by providing complete body support and restraint. The proper re-
straint system has three functions to perform: it must be comfortable under
normal conditions, it must fully restrain its occupant under the highest
expected g-loading, and it must attenuate rather than magnify accelerations
which are imposed by a vehicle upon its occupant(s).

Restraint comfort is mainly a function of position and pressure loading, and
it is most efficiently accomplished by "cut and try" development once the
optimum dynamic characteristics have been calculated and verified.

Acceleration transmission is a function of the resiliency between the ve-
hicle's structure and the occupant, their nonlinearities, and whether or
not they "bottom out". This resiliency is due to cushions (or net support),
restraint harness, flexibilities between the seat pan and the "rigid"
structure, and any means by which energy might be dissipated.

Without the addition of any intervening energy attenuator, impact energy is
transmitted directly through the rigid seat structure to the seat pan and
then applied to the occupant. The g-time profile actually experienced by
the occupant is influenced by the dynamic response of the occupant-restraint
system to the basic forcing function. Using the dynamic response factor, it
may be shown that a greater degree of restraint on the occupant results in
smaller inertia loads being transmitted.

The development of an optimum restraint-cushion system was not a primary
objective of this effort. However, a system was designed that will meet
most of the requirements as listed in the design criteria section. The
restraint harness consists of a lap belt, a shoulder harness, and a crotch
strap. All terminate in a single point where the actuation of a single
lever will release all straps simultaneously. The harness is fabricated of
dacron fabric meeting requirements of MIL-W-25361(USAF). In order to re-
duce elongation, two thicknesses were necessary for each stage. One com-
promise in the harness design was necessary. It was considered necessary
to use a dual inertia reel for the shoulder harness mounted at the top of
the seat back. Only one reel was found to be available that was suitable
for this application. Pacific Scientific Company submitted a proposed mod-
ification to one of their existing reels to make it adaptable to the seat
configuration. However, it is designed to use straps 1.75 inches wide

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instead of 2.0 inches wide, as presented in the AvSER recommendations.

Thick seat and back cushions are highly desirable from a comfort standpoint. However, it was concluded that their use would allow the occupant to have a velocity relative to the seat that would cause him to "bottom out" against the seat bucket and induce high peak decelerations into the occupant. This restraint would also become loose and allow additional movement relative to a desired fixed position. Therefore, the cushion used consists of thin comfort pads of polyurethane over the formed surfaces of the seat bottom and back.

BALLISTIC PROTECTION

Protection from ballistic attack is a problem of concern in any design that is to be used under combat conditions, but the problem is compounded in the case of aircraft by reason of stringent weight limitations, by severe space limitation, and by reason of the catastrophic and immediate effects which may ensue as a result of even relatively minor injuries.

Weight and space limitations dictate that means for crew protection be limited in expanse and thickness to the minimum. On the other hand, the potentially disastrous results of injury to (particularly) the pilot militate against any compromise in the degree of protection to be afforded. Approaches to resolve these somewhat contradictory requirements include: searching out materials which have high rates of ballistic resistance to weight, protective efficiency by proper placement and shaping of materials, and an assessment of the degree of protection offered by already existing aircraft structure or auxiliary components.

Several materials and types of armor, including steel, dual hardness steel, aluminum, titanium and hard-faced ceramic composites, were considered. Based on considerations of all factors involved, of which areal density, thickness, fabrication, and installation problems were of prime importance, it was concluded that ceramic composites were the optimum materials for this application. A ceramic-fiber glass material recently developed with an areal density of less than other armor materials was considered for this application.

Ballistic protection is provided for the occupant's torso. It consists of a removable shell around the seat. Several flat panels for the back sides, bottom, etc., are assembled to form the shell, which is supported at the points where the seat bucket attaches to its support structure. The armor is not used as structure. It is assumed that the occupant will wear a protective vest. Weight of this item is accounted for in the design analysis.

Details of the armor materials construction and ballistic protective capability are classified and are not included in this document. This information is contained in Reference 1.

INSTALLATION IN UH-1B HELICOPTER

One of the design requirements was that the crash seat be compatible with

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and adaptable to the UH-1B aircraft. The Hayes seat can be installed in
the UH-1B and does not compromise any space, control, ingress, and egress
beyond the present armored seat installations. Although the support struc-
ture is aft of the seat bucket, the structure does not protrude as far aft
into the cargo area as the present lap belt attachment to the floor (refer-
ence Figure 13).

In order to obtain compatibility with the UH-1 in an optimum manner that
would require minimum modification to the aircraft structure, the seat was
designed to be installed in a manner similar to the present seat. The track
spacing, location, and the track section are the same. The tracks are
mounted on top of the floor at BL 14 and BL 30 right hand and left hand.
Other locations would require substantially more reinforcement to the floor
structure. However, a wider track spacing would be advantageous for the
seat design and would result in slightly lower loads to the aircraft floor.

The floor modification consists primarily of reinforcing the floor beams at
BL 14 and BL 30, the cross frames between them, and the track attachment to
the beams. Available drawings of the floor structure show several different
versions of the floor structure. All are similar but vary in details. It
was impractical and impossible to design installations for all of these
various configurations under this contract effort. Therefore, the design
as developed shows only the basic reinforcements that are necessary. Detail
information for a specific group of aircraft could easily be developed from
the design information on the drawings. Appendix I contains a structural
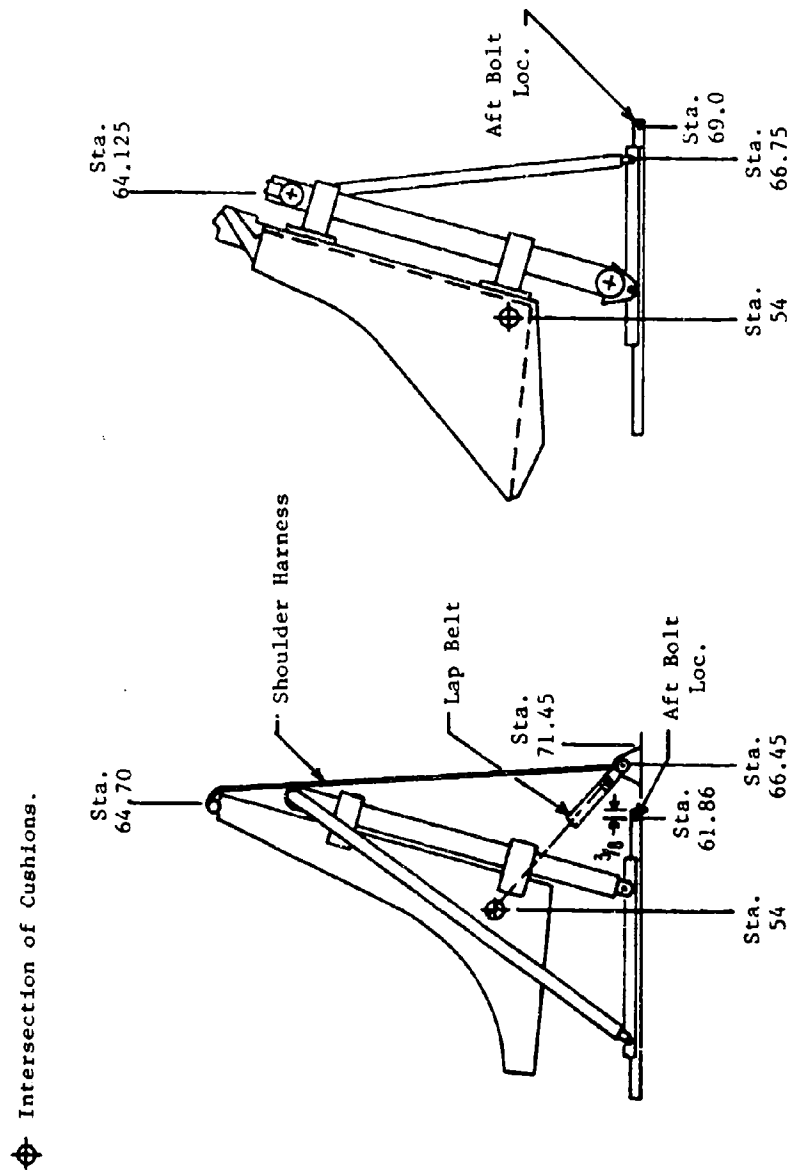analysis of the modification required.

Figure 13. Aft Positions of Present Seat and Hayes Seat in UH-1 Aircraft.

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TABLE V
WEIGHT SUMMARY OF CRASH SEAT

Item	Weight (lb.)
Bucket	24.80
Support Frame (includes the vertical and forward energy absorption devices)	37.54
Vertical Slide Assembly (includes the lateral energy absorption devices)	11.51
Vertical Actuator	6.00
Inertia Reel	4.70
Restraint Harness	3.16
Cushions	0.99
Weight of Seat	88.70
Armor	
Mounted to Bucket	95.00
Vest Type Chest Armor (GFE)	14.00
Weight of Armor	109.00
Total Seat and Armor Weight	197.70

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CONCLUSIONS

The following conclusions are made based on the study of the four seats and the design of a new improved seat.

1. Each of the four seat designs tested allowed loads on the occupant to exceed the design limit and the published human tolerances.

2. The series of tests did not adequately test the several concepts of energy absorption.

3. Seats designed to the criteria used for the four seats will not provide adequate crash protection for the occupant.

4. Restraint systems meeting current military specifications as used on the seats investigated are inadequate to restrain the occupant in a crash environment.

5. The revised design criteria presented herein and used for the design of a new improved seat are considered to be reasonable.

6. The seat designed under this contract represents an optimum approach to meeting all specified requirements.

7. Design criteria and the effectiveness of crash and ballistic protection characteristics can best be determined by fully testing the new seat.

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RECOMMENDATIONS

On the basis of the foregoing conclusions, the following recommendations are made.

1. The aircrew crash survival seat designed under this contract should be completely tested to determine its compatibility with design requirements. The tests should include static and dynamic load tests, and ballistic tests.

2. Further work should be directed toward the development of an optimum restraint and cushion system.

3. The definition of a design acceleration spectrum or input to the seat in a crash environment should be further defined.

4. A mathematical analysis should be made of the dynamics of the seat, occupant, and seat support system to determine the response to the input acceleration.

5. Consideration should be given to the development of an armored crash survival seat design in which the armor is an integral part of the structure. This could result in a reduction of total weight.

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REFERENCE DRAWINGS

Drawing Number	Title
201-00001	Seat Assembly
201-00002	Structure Assembly
201-00003	Vertical Support Weldment
201-00006	Fitting, Horizontal Slide, Upper
201-00007	Vertical, Attenuator Attach Tube
201-00008	Horizontal Attenuator Tube
201-00009	Carriage, Lower
201-00011	Aft Support
201-00012	Rod Assembly
201-00013	Upper Support Tube
201-00016	Fitting, Aft Tube, Lower
201-00019	Pin, Adjust Assembly
201-00021	Bottom Linkage Assembly
201-00024	Guide - Horizontal Adjust
201-00027	Rod - Horizontal Adjust
201-00029	Fitting, Horizontal Slide, Lower
201-00031	Armor Assembly
201-00032	Bucket Assembly
201-00033	Tube, Actuator Attach, Lower
201-00035	Track, Seat
201-00036	Seat Installation
201-00038	Tube, Crotch Strap Support
201-00039	Clevis, Lap Belt End Fitting
201-00042	Stiffener, Bottom Skin

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201-00045	Fitting, Lap Belt Clevis Pivot
201-00047	Support, Inertia Reel Assembly
201-00051	Arm, Inertia Reel Support
201-00071	Pin, Forward Roller

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APPENDIX I. STRESS ANALYSIS

An analysis of the seat loads and load distribution, based on the structural criteria outlined in the Design Criteria section, and the stress analysis of major components are presented in this appendix.

LOAD ANALYSIS

The design criteria represent, in static terms, load and energy absorber requirements for which the seat must be designed to insure that the occupant does not sustain loads greater than recognized human tolerances under dynamic load conditions and that the seat does not fail or become disengaged from the structure in a survivable crash of a helicopter. These requirements are summarized in the following four design conditions:

1. A forward load along the longitudinal axis of the helicopter of not less than 22g and not exceeding 35g (based on forward attenuation).

2. A vertical load along the vertical axis of the helicopter of $17\pm2g$ based on 80 percent of the occupant's weight.

3. A lateral load applied parallel to the lateral axis of the helicopter of $4,000\pm400$ pounds total.

4. A combined load consisting of loads of the following ratios: 75 percent of the forward load, 100 percent of the vertical load, and 50 percent of the lateral load.

It is required on the first three conditions stated above that attenuation of the occupant and seat be provided at the load levels prescribed. The minimum forward loading is determined by the distance over which the load can be attenuated as per Figure 9.

The seat is so designed that its forward load deflection curve carries into the shaded region above curve B without first falling below the base curve, A. This is accomplished by providing a forward attenuation over 5.62 inches which results in a minimum load of 22.5g. Thus, to meet the range of occupant weight and full and no-armor conditions, the maximum total weight (200-pound occupant, seat, and full armor) is applied to the structure of 22.5g.

$$P_{FWD} = 22.5 \text{ (Pilot Wt. + Armor Wt. + Seat Wt. + Support Structure Wt.)}$$

For

Pilot Wt. = 200 lb.

Armor Wt. = 95 lb. + 14 lb. (Vest) = 109 lb.

Seat and Support Structure Wt. = 85 lb.
(Based on preliminary weights calculations)

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$$P_{FWD} = 2?.5 \ (200 + 109 + 85) = 22.5(394) = 8,860 \ \text{lb}.$$

For

Pilot Wt. = 200 lb.

Armor Wt. = 0

Seat and Support Structure Wt. = 85 lb.

$$P_{FWD} = 22.5 \ (200 + 85) = 22.5(285) = 6,410 \ \text{lb}.$$

Therefore, when the armor is used, the maximum deceleration load that the 135-pound occupant will feel is

$$g = 8,860/(135 + 109 + 85) = 25.9$$

A separate set of attenuators is used for the seat without armor. The total forward load is 6,410 pounds, and the maximum deceleration load that the 135-pound occupant will feel is

$$g = 6,410/(135 + 85) = 29.1$$

An analysis of the restraint system, seat bucket, and harness was attempted in order to substantiate the requirements set forth in Reference 19. Various parameters such as occupant size and position, harness strap(s) orientation, and bucket shape were examined in order to obtain the most suitable restraint system. From this preliminary analysis, the restraint system presented was concluded to be the most suitable. However, due to the inherent problems associated with reacting loads through a pliable medium such as the human body, it is felt that only tests conducted under the proper guidance can accurately determine a restraint system load distribution. Therefore, the complete analysis is not presented herein except as it pertains to the load requirements specified in Reference 19.

CONDITION 1 - FORWARD LOAD

For this forward loading condition, the occupant is considered to be completely restrained by the harness system. This loading is depicted in Figure 14.

$$P_X = 22.5 \ (220 + 14) = 4,820 \ \text{lb}.$$

where

Weight of occupant = 200 lb.

Weight of chest armor = 14 lb.

The harness loads are conservatively assumed to meet the harness load requirements specified in Table IV as follows:

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C(crotch strap) = 2,200 lb.

S(shoulder strap) = 3,600 lb. (2 straps)

L(lap belt) = 5,000 lb. (loop)
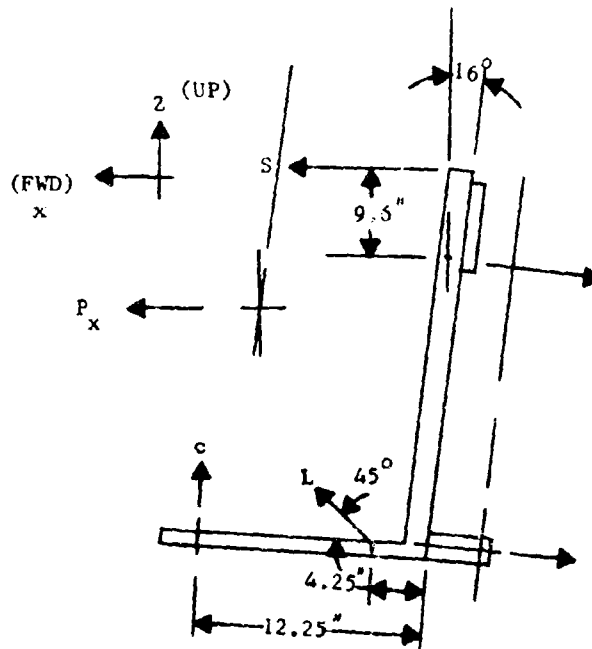
Figure 14. Seat, Top Adjustment Position, Condition 1.

CONDITION 2 - VERTICAL LOAD

For this vertical loading condition, the occupant is restrained completely by primary seat structure, the bottom and back, as depicted in Figure 15.

$$P_z \quad 17 \left[200(0.80) + 14 \right] = 2,960 \text{ lb.}$$

where

80 percent of the occupant's weight is assumed to be reacted by the seat = 200 x .80 = 160 lb.

$$\Sigma F_x = 0$$

$$R_z(\sin 5^\circ) = R_x (\cos 16^\circ)$$

$$R_x = (0.0871/0.961) R_z = 0.0906(R_z)$$

$$\Sigma F_z = 0$$

$$R_z(\cos 5°) + R_x (\sin 16°) = 2,960 \text{ lb.}$$

$$0.996(R_z) + 0.276(0.0906R_z) = 2,960 \text{ lb.}$$

$$R_z = 2,960/1.021 = 2,900 \text{ lb.}$$

$$R_x = 0.0906(2,900) = 263 \text{ lb.}$$
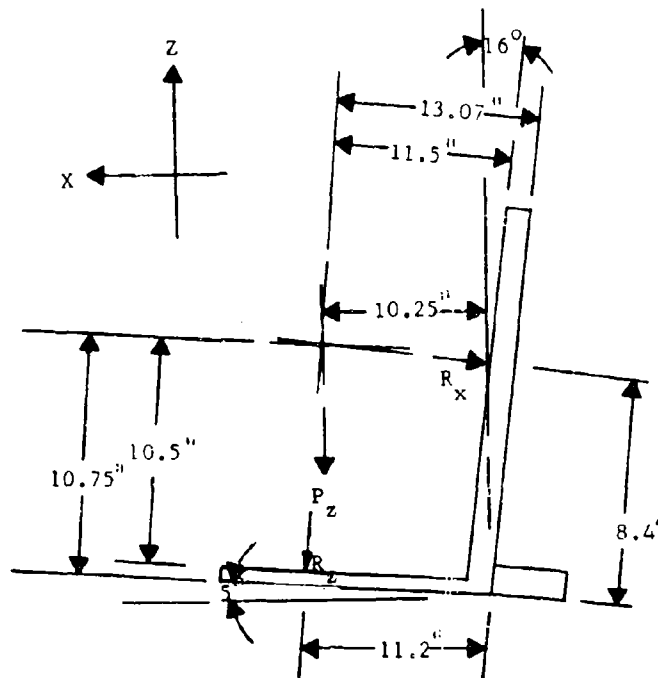
Figure 15.   Seat, Top Adjust Position, Condition 2.

CONDITION 3 - LATERAL LOAD

For the lateral loading condition, it is assumed that the occupant is completely restrained by the side skin of the seat. This is accomplished by the occupant's sliding to one side, approximately 0.9 inch, until leg contact is made with the side skin. Therefore, in this attitude the occupant bears out in the skin at his leg, hip and rib-cage positions. Also, an overlapping assumption is made that 50 percent of the total side load is resisted by the lap belt in tension and the occupant's bearing against the

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vertical post on the downwind side.
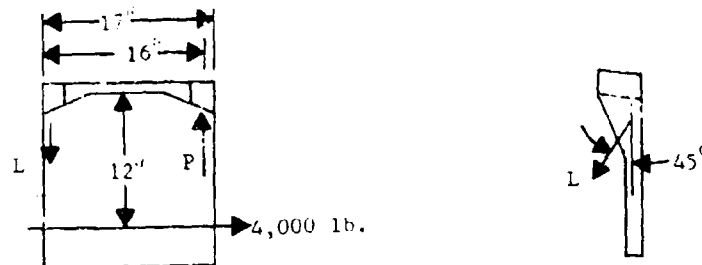
$$0.707(L) = 50 \times 4000 \frac{(12)}{16} = 1,500 \text{ lb.}$$

$$L = 2,120 \text{ lb.}$$

CONDITION 4 - COMBINED LOADS

For combined loading, the deceleration loads applied to the seat occupant are

$$\text{Forward load, } P_x = 0.75 \times 4810 = 3,610 \text{ lb.}$$

$$\text{Vertical load, } P_z = 1.00 \times 2960 = 2,960 \text{ lb.}$$

$$\text{Lateral load, } P_y = 0.50 \times 4,000 = 2,000 \text{ lb.}$$

Utilizing the composite seat structure and harness system, these applied loads are reacted separately in the same manner as their respective independently applied loads. This is shown in Figure 16 wherein the harness and seat bucket loads are assumed as follows:

$$C = 0.75 \times 2,200 = 1,650 \text{ lb.}$$

$$S = 0.75 \times 3,600 = 2,700 \text{ lb. (2 straps)}$$

$$L = 0.75 \times 5,000 = 3,750 \text{ lb. (loop)}$$

$$R_z = 2,900 \text{ lb.}$$

$$R_x = 0 \text{ lb. (due to forward load)}$$

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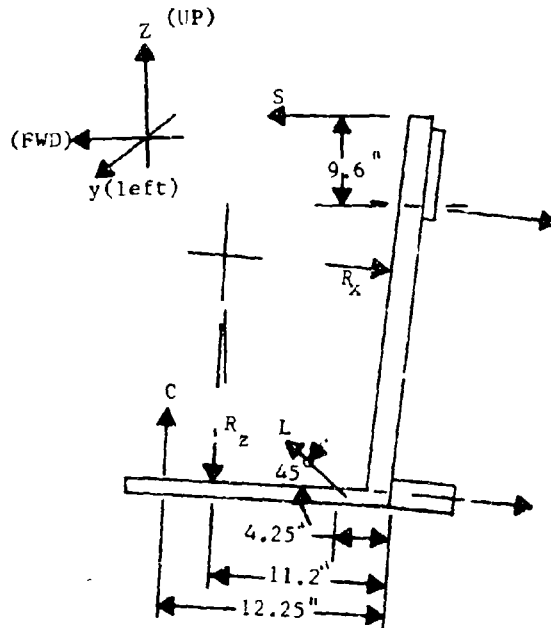
Figure 16. Seat, Top Adjust Position, Condition 4.

SUPPORT STRUCTURE LOADS

The support structure, reference Figure 17, is primarily a truss utilizing thin-wall steel tubing to obtain the best strength-to-weight ratio. Various members comprising the structure transmit loads by bending action in addition to their axial requirements. The support structure basic design is controlled by the energy absorption strokes, forward, vertical, and lateral, that necessitate certain clearances between the seat and the UH-1B helicopter interior. This is especially true for the armored seat configuration such that the lateral attenuation of 2 inches requires that there be no support structure along the side of the seat. To provide sufficient vertical attenuation, all seat-to-support structure attachments are placed on the seat back and not along the seat bottom. These considerations thus necessitate a seat support structure that is located entirely aft of the seat bucket.

To obtain the loads applied to the support structure, the deceleration load limits are applied at the center of gravity of the occupant, seat and armor. The occupant c.g. is that specified in Reference 17; seat and armor weight and c.g. locations were obtained from preliminary weight calculations. For load conditions involving a laterally applied load, the occupant is assumed to slide laterally until body contact is made with the seat side skin, approximately 0.9 inch (reference page 54).

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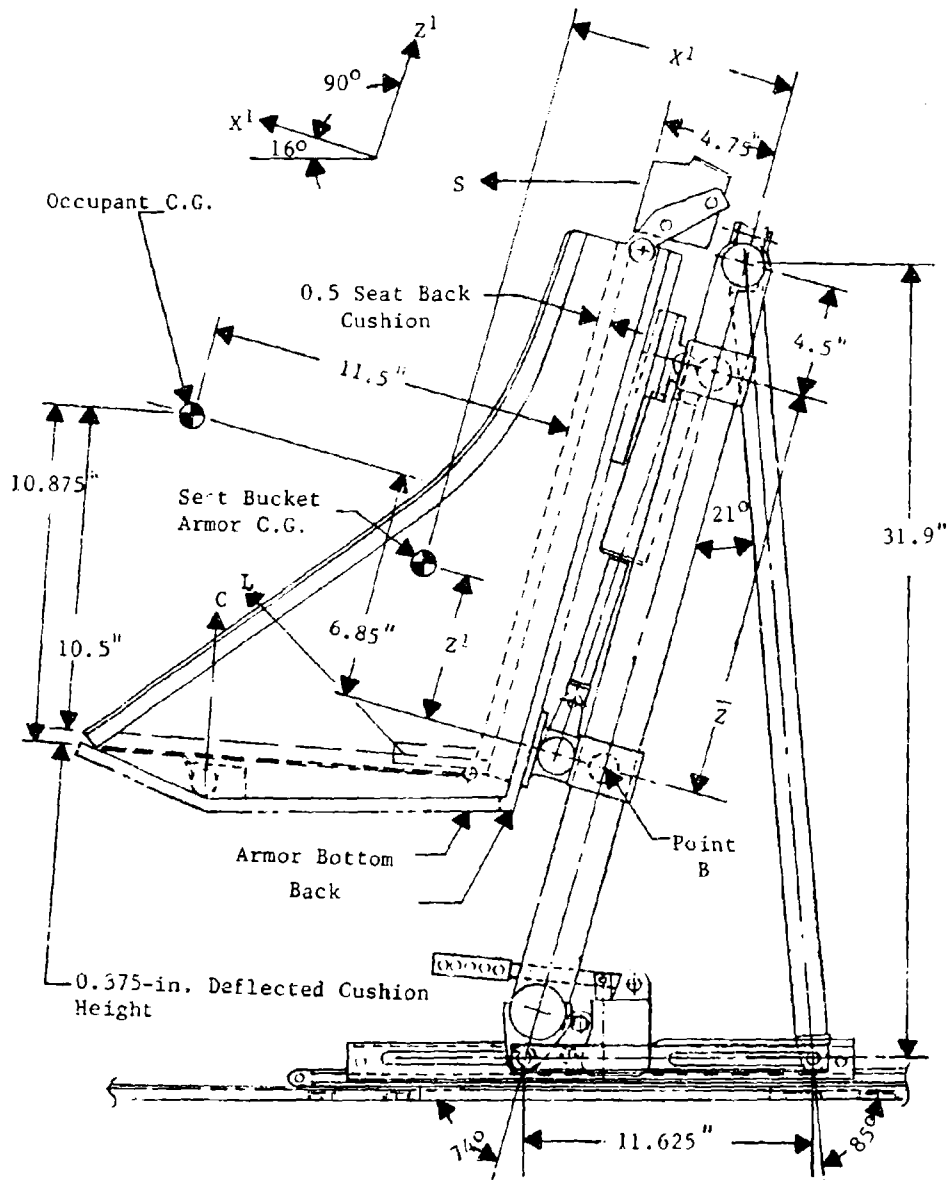
(SIDE ARMOR IS OMITTED FOR CLARITY)

Figure 17. Seat Dimensional Data.

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TABLE VI
SEAT AND OCCUPANT CENTER-OF-GRAVITY LOCATIONS

Item[1]	Weight (lb.)	Top Adjust Position $\bar{Z} = 14.125$ in.		Center Adjust Position $\bar{Z} = 16.625$ in.	
		X^1 (in.)	Z^1 (in.)	X^1 (in.)	Z^1 (in.)
Seat Bucket[2]	50.0	3.17	9.35	3.17	9.95
Armor[3]	95.0	5.69	8.75	5.69	8.75
Seat Bucket plus Armor	145.0	4.84	8.95	4.84	9.16

[1] Reference Figure 17 for X^1 and Z^1 dimensions.

[2] Includes bucket, actuator and sliding structure for vertical attenuation.

[3] Does not include chest armor.

A summary of internal loads for the support structure is presented in Table VII for all four design conditions. The 200-pound occupant weight is used in the seat top adjustment position, as this is the critical design position. Table VIII presents the attenuator requirements for each of the three independent loading conditions for the seat center adjustment position with the 200-pound and 135-pound occupants. This is done to best accomplish load attenuation under all adjustment positions utilizing armor and no-armor configurations.

All load conditions except the lateral are critical for the full armor configuration. Sample calculations are presented for Condition 1, Forward Loading.

$$P_1 = 22.5(200 + 14) = 4,820 \text{ lb.}$$

$$P_2 = 22.5(145) = 3,260 \text{ lb.}$$

The sliding surfaces are coated with Electrofilm Lubricant No. 66-C. Assume that a coefficient of friction of $\mu = 0.1$ is developed between the vertical slide fittings and the front support legs and that, for this symmetrical load condition, $R_{UL} = R_{UR}$ and $R_{LL} = R_{LR}$.

TABLE VII
SUMMARY OF SUPPORT STRUCTURE LOADS

Seat-Top Adjust Position	Condition 1 Forward Loading (lb.)	Condition 2[1] Vertical Loading (lb.)	Condition 3[2] Lateral Loading (lb.)	Condition 4 Combined Loading (lb.)	Full Fwd, Full Side (lb.)
R_{UL}	2,747	703	-1,440	2,520	-
R_{LL}	893	-2,197	-1,480	-2,652	-
R_{UR}	2,747	703	+1,440	3,430	-
R_{LR}	893	-2,197	+1,480	-1,208	-
R_{YU}	-	-	1,040	688	-
R_{YL}	-	-	960	312	-
A_1	1,579	4,864	-	6,700	-
R_x^1	483	1,490	-	2,050	-
R_s^1	1,500	4,630	-	6,370	-
R_1	2,760	- 350	1,890	2,434	-
R_2	877	-1,145	1,031	- 214	-
R_3	2,760	- 350	-1,890	1,015	-
R_4	877	-1,145	-1,031	-1,142	-
R_1^1	3,002	+ 395	2,272	+3,601	-
R_3^1	3,002	+ 395	-2,272	1,906	-
P_3	8,386	1,100	6,350	10,060	-
P_4	8,386	1,100	-6,350	5,310	-
P_1^1	-7,820	-1,028	-5,920	-9,390	-
P_2^1	-7,820	-1,028	+5,920	-4,960	-
P_1	-8,570	-3,340	-5,022	-12,201	-
P_2	-8,570	-3,340	+5,022	-8,514	-
P_{11}^1	-8,934	-3,630	-5,022	-12,875	-
P_{11}	-8,934	-3,630	-6,052	-13,259	-13,142
P_{22}^1	-8,934	-3,630	+5,022	-9,131	-
P_{22}	-8,934	-3,630	+6,052	-8,750	-4,728
R_2^1	877	-1,145	1,244	- 119	1,763
R_4^1	877	-1,145	-1,244	-1,238	- 9
R_{Y1}	-	-	1,318	730	1,340
R_{Y2}	-	-	683	269	659
P_5	8,386	1,100	6,350	10,060	-
P_6	8,386	1,100	-11,600	+2,400	-
P_7	-	-	5,900	3,270	-

TABLE VII (CONTINUED)

Seat-Top Adjust Position	Condition 1 Forward Loading (lb.)	Condition 2[1] Vertical Loading (lb.)	Condition 3[2] Lateral Loading (lb.)	Condition 4 Combined Loading (lb.)	Full Fwd, Full Side (lb.)
P_8	-	-	0	0	-
P_9	8,386	1,100	11,600	+12,970	18,358
P_{10}	8,386	1,100	-11,600	+ 2,400	- 598

[1] Based on 80% of occupant weight

[2] No-armor configuration critical

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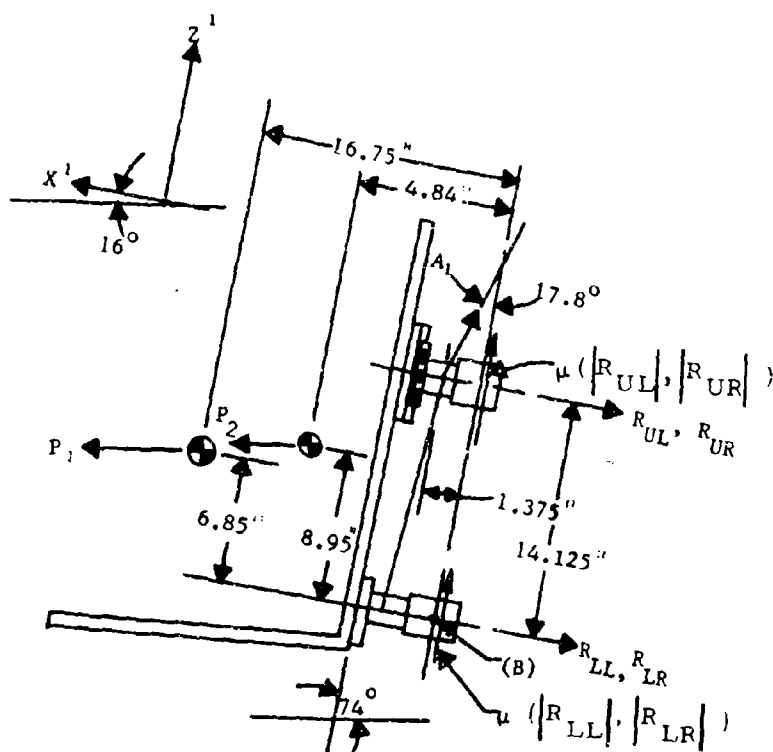
Figure 18. Example of Load Diagram.

$\Sigma F_z^1 = 0$

$(\cos 17.8^\circ) A_1 + 0.1 (2R_{UL}) + 0.1 (2R_{LL}) = (P_1 + P_2) \sin 16^\circ$

$0.952(A_1) + 0.2(R_{UL}) + 0.2(R_{LL}) = 8,080(0.276) = 2,230 \qquad (1)$

$\Sigma F_x^1 = 0$

$(\sin 17.8^\circ) A_1 + 2(R_{UL}) + 2(R_{LL}) = (P_1 + P_2) \cos 16^\circ$

$0.306(A_1) + 2(R_{UL}) + 2(R_{LL}) = 8,080 (.961) = 7,770 \qquad (2)$

$\Sigma M_B = 0$

$(\cos 17.8^\circ) \ (A_1) \ 1.375 + (\sin 17.8^\circ) \ A_1 \ (14.125) + 2(R_{UL}) \ 14.125$

$= (\cos 16^\circ) P_1 (6.85) + (\sin 16^\circ) \ P_1 (16.75) + (\cos 16^\circ) \ P_2 (8.95)$

$+ (\sin 16^\circ) \ P_2 (4.84)$

$5.62 \ (A_1) + 28.25 (R_{UL}) = 11.21 \ (4,820) + 9.93 \ (3,260)$

$R_{UL} = 1,915 + 1,145 - 0.1985 \ (A_1) = 3,060 - 0.1985 \ (A_1)$ (3)

Substituting equation 1 into equation 2,

$R_{LL} = 11,150 - (R_{UL}) - 4.76 \ (A_1)$

$0.306(A_1) + 2(R_{UL}) + 2 \left[11,150 - (R_{UL}) - 4.76 \ (A_1) \right] = 7,770$

$-9.214(A_1) = -22,300 + 7,770 = -14,530$

$A_1 = 1,579 \ \text{lb.}$

$R_{UL} = 3,060 - 0.1985(1,579) = 2,747 \ \text{lb.}$

$R_{LL} = 11,150 - 2,747 - 4.76(1,579) = 893 \ \text{lb.}$

$R_{UL} = 2,747 \ \text{lb.} = R_{UL}$

$R_{LL} = 893 \ \text{lb.} - R_{LR}$

$A_1 = 1,579 \ \text{lb.}$

For loads applied to the support structure,

$R_X^{\ 1} = A_1 \ (\sin 17.8^\circ) = 1,579 \ (0.308) = 483 \ \text{lb.}$

$R_z^{\ 1} = A_1 \ (\cos 17.8^\circ) = 1,579 \ (0.952) = 1,500 \ \text{lb.}$

Equations for Support Structure Member Loads (Reference Figure 20)

$R_1 = .864 \ R_{UR} + .436 \ R_{LR}$

$R_2 = .136 \ R_{UR} + .564 \ R_{LR}$

$R_3 = .864 \ R_{UL} + .436 \ R_{LL}$

$$R_4 = .136 R_{UL} + .564 R_{LL}$$

$$R_1^1 = .952 R_{UR} + .480 R_{LR} - .088 R_{UL} - .044 R_{LL} + .5 R_X^1$$

$$R_3^1 = .952 R_{UL} + .480 R_{LL} - .088 R_{UR} - .044 R_{LR} + .5 R_X^1$$

$$P_3 = 2.656 R_{UR} + 1.339 R_{LR} - .245 R_{UL} - .123 R_{LL} + 1.395 R_X^1$$

$$P_4 = 2.656 R_{UL} + 1.339 R_{LL} - .245 R_{UR} - .123 R_{LR} + 1.395 R_X^1$$

$$P_1^1 = -\left[2.481 R_{UR} + 1.251 R_{LR} - .229 R_{UL} - .115 R_{LL} + 1.303 R_X^1\right]$$

$$P_2^1 = -\left[2.481 R_{UL} + 1.251 R_{LL} - .229 R_{UR} - .115 R_{LR} + 1.303 R_X^1\right]$$

$$P_1 = -\left[2.253 R_{UR} + 1.136 R_{LR} + 1.303 R_X^1 + .5 R_z^1\right]$$

$$P_2 = -\left[2.253 R_{UL} + 1.136 R_{LL} + 1.303 R_X^1 + .5 R_z^1\right]$$

$$P_{11} = -\left[2.483 R_{UR} + 1.252 R_{LR} + 1.303 R_X^1 + .5 R_z^1 - .230 R_{UL} - .116 R_{LL}\right]$$

$$P_{22} = -\left[2.483 R_{UL} + 1.252 R_{LL} + 1.303 R_X^1 + .5 R_z^1 - .230 R_{UR} - .116 R_{LR}\right]$$

$$P_{11}^1 = -\left\{2.253 R_{UR} + 1.136 R_{LR} + 1.303 R_X^1 + 0.5 R_z^1 + 0.1 \left[|R_{UR}| + |R_{LR}| + |R_{YU}| + |R_{YL}|\right]\right\}$$

$$P_{22}^1 = -\left\{2.253 R_{UL} + 1.136 R_{LL} + 1.303 R_X^1 + 0.5 R_z^1 + 0.1 \left[|R_{UL}| + |R_{LL}| + |R_{YU}| + |R_{YL}|\right]\right\}$$

The two previous equations are not applicable for the independent lateral load condition; for the lateral loading, $P_{11}^1 = P_1$ and $P_{22}^1 = P_2$.

$$R_2^1 = .150 R_{UR} + .622 R_{LR} - .014 R_{UL} - .058 R_{LL}$$

$$R_4^1 = .150 R_{UL} + .622 R_{LL} - .014 R_{UR} - .058 R_{LR}$$

$$R_{Y_1}^{1} = .864 \; R_{YU} + .436 \; R_{YL}$$

$$R_{Y_2} = .136 \; R_{YU} + .564 \; R_{YL}$$

Loads for Lateral Acceleration to Left Only

$$P_7 = 4.48 \; R_{Y_1}$$

$$P_8 = 0$$

$$P_5 = P_3$$

$$P_6 = P_4 - 3.99 \; R_{Y_1}$$

$$P_9 = P_3 + 3.99 \; R_{Y_1}$$

$$P_{10} = P_4 - 3.99 \; R_{Y_1}$$

TABLE VIII

SUMMARY OF ATTENUATOR LOAD REQUIREMENTS

		Total Attenuator Requirements						
		Condition 1 Forward Loading		Condition 2 Vertical Loading[1]		Condition 3 Lateral Loading		
		g	lb.	g	lb.	g	lb.	
	Seat-Center Ad-just Position	Load		Load		Load	Upper Fitting	Lower Fitting
A.	Armored Seat Configuration							
	1. 200-lb. occupant	22.5	8,505	17	4,960	11.15	1,400	1,537
	2. 135-lb. occupant	25.9	8,505	17	4,355	13.60	1,493	1,547
B.	No-Armor Configuration							
	1. 200-lb. occupant	22.5	6,160	17	3,360	16	1,276	1,473
	2. 135-lb. occupant	29.1	6,160	17	2,630	21.60	1,345	1,472

(1) Loads are based on 80 percent of the occupant weight. As the seat moves vertically, the angle that the attenuators make with the front support legs will decrease, resulting in an increase in deceleration loading of approximately 1g.
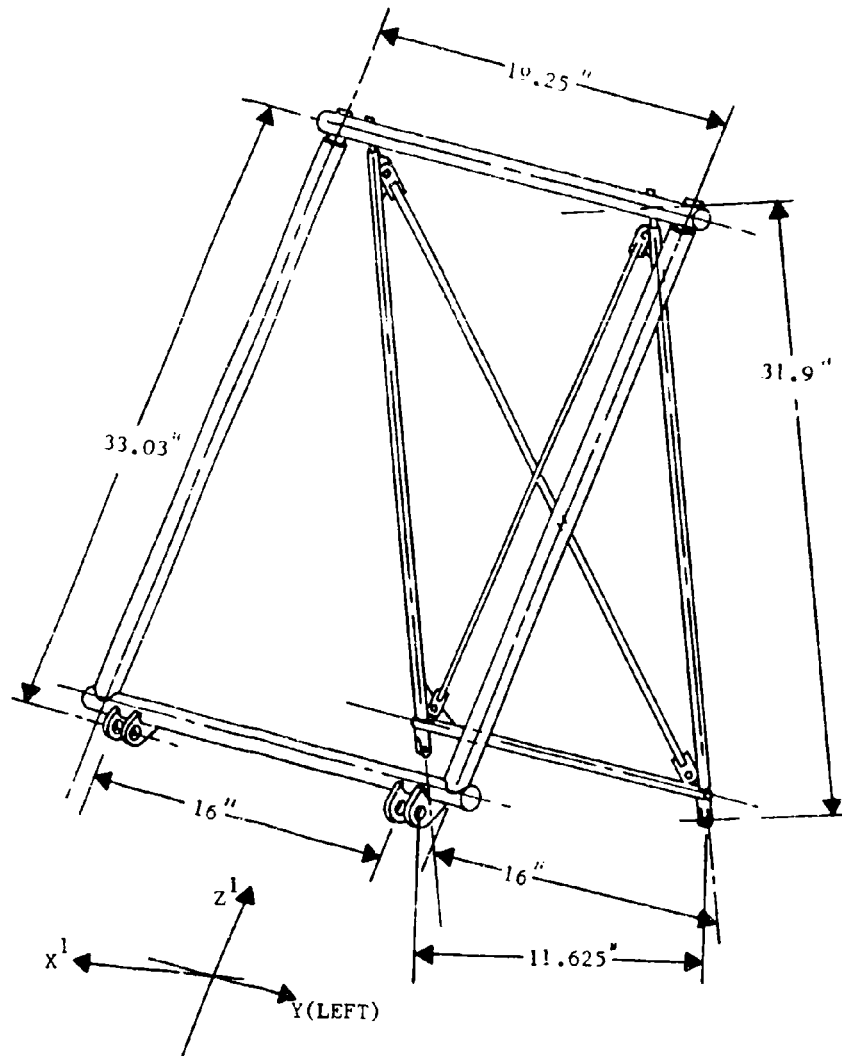
Figure 19. Support Structure.

Figure 20. Support Structure Load Diagram.

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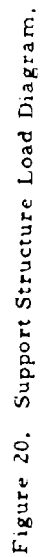
SEAT BUCKET FRAME (Reference Drawing 201-00032)

The bucket frame is composed of two back beams, to which the support struc-
ture attachments are mounted, and two bottom beams, which are cantilevered
off the back beams (see Figure 17). For applied loads tending to rotate
the seat bottom lip up, the structure works as individual beams; however,
for applied loads tending to rotate the bucket lip down, the beams are
supported by the side skins. The forward load condition, seat top adjust
position, has been found by preliminary analysis to be the design condition
for both the bottom and back beams.
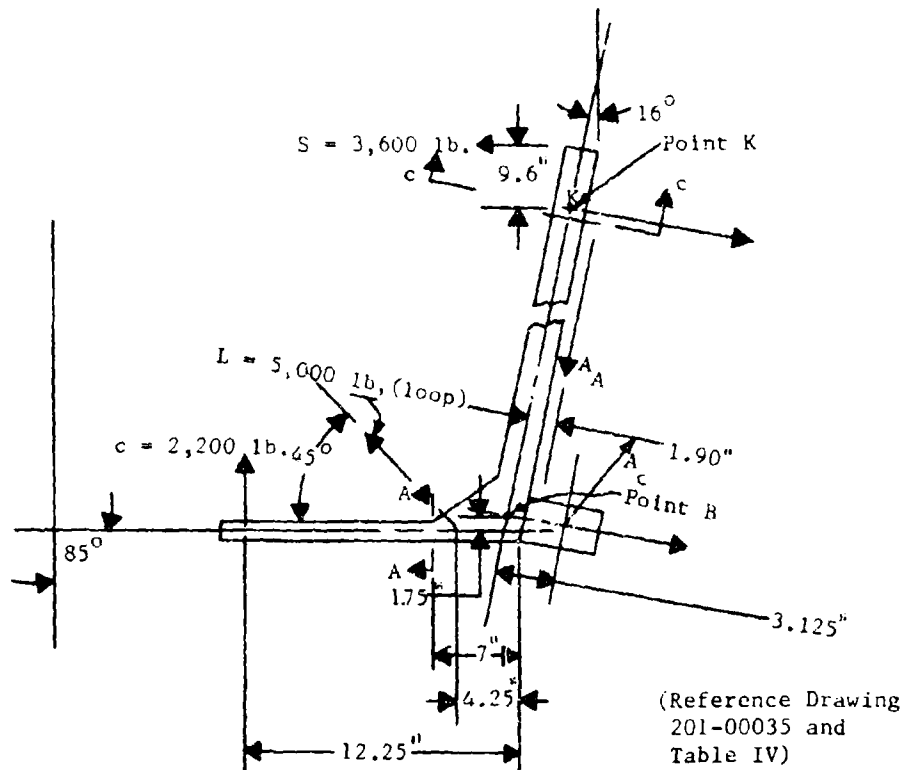
Figure 21. Seat Bucket Loads.

INERTIA REEL ATTACHMENT (Reference Drawing 201-00027)

The Pacific Scientific Inertia Reel type is designed for 4,000 pounds com-
bined total strap load per manufacturer's drawing 0107110-1. Therefore,
all reel attachment points are adequate for the design strap load of 1,800
pounds, and only attachment structure will be analyzed.
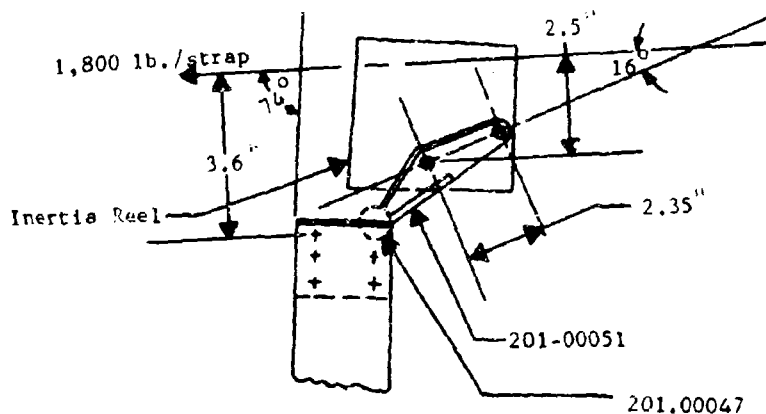
All margins of safety for the reel attachment are greater than + .25 except those analyzed herein.

INERTIA REEL SUPPORT TUBE - PART NUMBER 201-00027
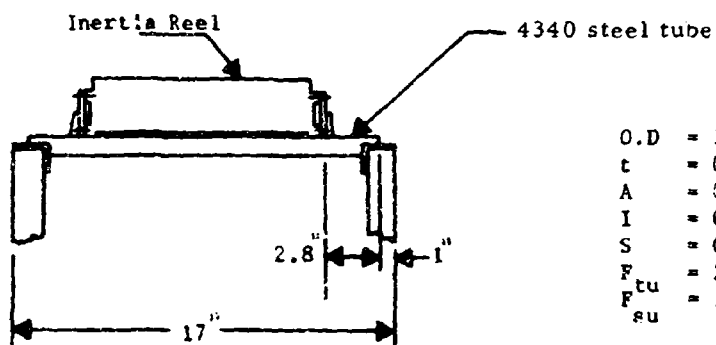
$$O.D = 1.0 \text{ in.}$$
$$t = 0.065 \text{ in.}$$
$$A = 0.191 \text{ in.}^2$$
$$I = 0.021 \text{ in.}^4$$
$$S = 0.04193 \text{ in.}^3$$
$$F_{tu} = 200,000 \text{ p.s.i.}$$
$$F_{su} = 119,000 \text{ p.s.i.}$$

$$T = 1,800 \ (3.6) = 6,470 \text{ in.-lb.}$$

$$M = 1,800 \ (2.8) = 5,050 \text{ in.-lb.}$$

$$f_{bu} = \frac{5,050}{.04193} = 120,500 \text{ p.s.i.}$$

$$f_{stu} = \frac{6,470(0.5)}{2(0.021)} = 77,000 \text{ p.s.i.}$$

$$f_{su} = \frac{2(1,800)}{0.191} = 18,860 \text{ p.s.i.}$$

For $L/_{\rho} = \frac{15}{1} = 15$, $D/_t = 15.39$

$F_{stu} = 112,000$ p.s.i.

$F_{bu} = 263,000$ p.s.i.

$$M.S. = \frac{1}{\sqrt{R_{st}^2 + R_{b}^2 + R_{s}}} -1$$

where

$$R_{st} = \frac{f_{stu}}{F_{stu}} = \frac{77,000}{112,000} = 0.689$$

$$R_{b} = \frac{f_{bu}}{F_{bu}} = \frac{120,500}{263,000} = 0.458$$

$$R_{s} = \frac{f_{su}}{F_{su}} = \frac{18,860}{119,000} = 0.1584$$

$$\therefore M.S. = \frac{1}{\sqrt{.684 + 0.1584}} -1 = + 0.01$$

The maximum fastener load from the inertia reel obtained by conventional multiple fastener load distribution analysis is 1,066 pounds. These fasteners must also transmit the track loads to the back beams for the seat in the bottom adjust position. The two loads are on different shear planes, but they both combine and bear out in the back beams. The track attachment loads for the bottom adjust position are less by observation than those for the top adjust position. For brevity of analysis, the load for the top adjust position is conservatively added to the fastener load from the reel attachment. This track attachment load is determined by preliminary analysis to be 631 pounds.

$$P_{bru}^{1} (in.\ 0.05 + 0.032\ 2024-T42) = 1.12(1500)\frac{0.082}{0.080} = 1720\ lb.$$

(Reference 7)

$$M.S. = \frac{P_{bru}^{1}}{P_{bru}} -1 = \frac{1720}{1066 + 631} -1 = + 0.01$$

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VERTICAL BEAMS (Reference Drawing 201-00032)

All Sheet 2024-T42 Clad

A = 0.8048 in.^{2} I_x = 0.911 in.^{4}

$\overline{X}$ = 0 in. I_y = + .490 in.^{4}

$\overline{Y}$ = 1.43 in. I_{xy} = 0.107 in.^{4}
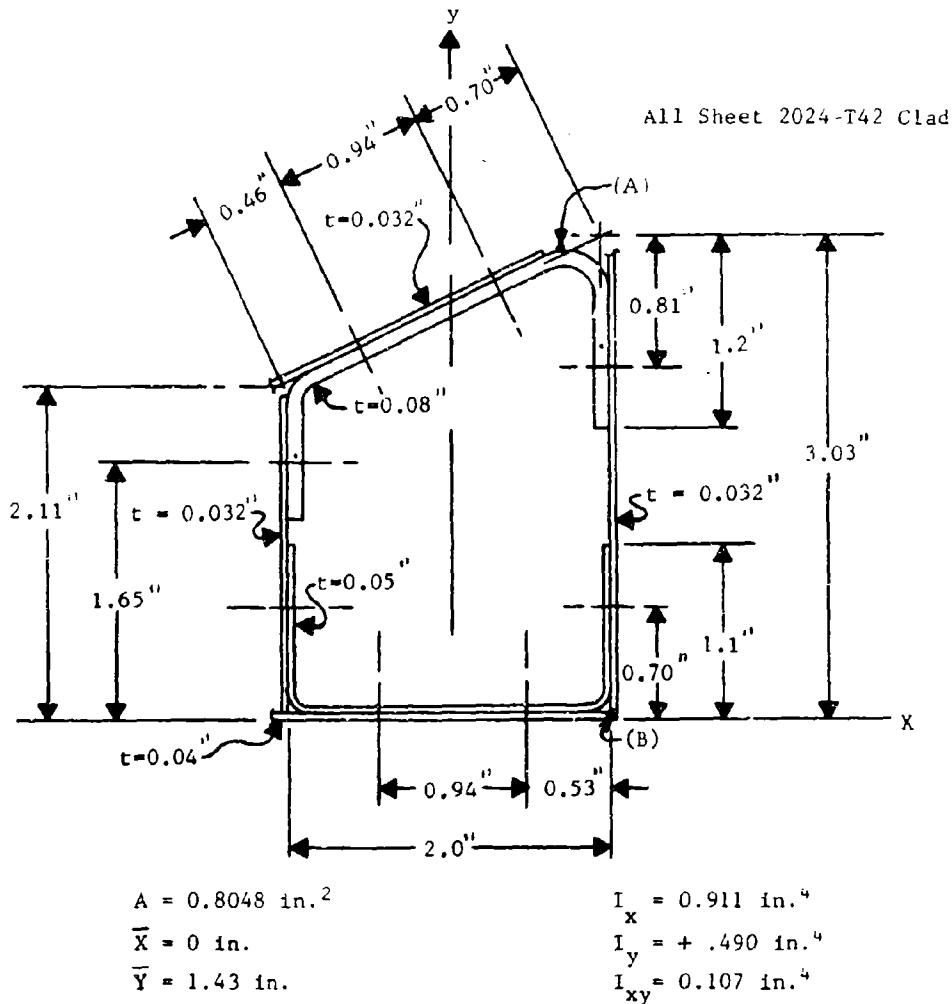
SECTION C-C

Figure 22.   Vertical Beams, Seat

The maximum bending loads on the aft beam are as follows (see Figure 2i for symbols and dimensions):

A_A = ½ (armor weight) (22.5)(sin 16°) assuming that one-half of the armor weight is reacted at the lower fitting.

A_A = ½ (95) 22.5 (0.276) = 295 lb.

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A_c = (pilot weight + armor weight + bucket weight) (sin 18.78°) 22.5 where the bucket weight includes all structure that moves with the bucket requiring vertical attenuation.

= (200 + 95 + 14 + 50) (0.323) (22.5) = 2,600 lb.

ΣM_K = 3,600 (9.6) = 34,500 in.-lb.

ΣM_B = 2,200 (12.25 − 1.5) + 0.707 (5,000) (4.25 − 1.5)

−2,600 (3.125) (cos 2.78°) + 295 (1.90)

= 23,400 + 9,730 − 8,130 + 561 = 25,561 in.-lb.

For the no-armor configuration,

A_A = 0

A_c = (200 + 50) (0.323) 22.5 = 1.820 lb.

ΣM_K = 34,500 in.-lb.

ΣM_B = 23,400 + 9,730 − 1,820 (3.125) (cos 2.78°) = 27,450 in.-lb.

f_b = −($K_3 M_y$ − $K_1 M_x$) x −($K_2 M_x$ − $K_1 M_y$) y

For M_x = $\dfrac{34,500}{2}$ = 17,750 in.-lb. each beam,

M_y = 0

P = $\dfrac{-3,600}{2}$(sin 16°) = −485 lb.

f_{bu} = M_x (K_1 x − K_2 y)

Point A, x = 0.8 −0 = 0.8 in., y = 2.95 −1.43 = 1.52 in.

Point B, x = 1.032 −0 = 1.032 in., y = −1.43 in.

$$K_1 = \frac{I_{xy}}{(I_x I_y - I_{xy}^2)} = 0.246 \ \frac{1}{in.^4}$$

$$K_2 = \frac{I_y}{(I_x I_y - I_{xy}^2)} = 1.125 \ \frac{1}{in.^4}$$

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$$f_{bu(A)} = 17,750 \left[0.2462(0.80) - 1.125(1.52) \right] = 17,750(-1.51) = 26,800 \text{ p.s.i.}$$

$$f_{bu(B)} = 17,750 \left[0.2462(1.032) - 1.125(-1.41) \right] = 17,750(1.865) = 33,100 \text{ p.s.i.}$$

$$f_c = \frac{P}{A} = \frac{485}{0.8048} = 604 \text{ p.s.i.}$$

For crippling analysis, assume an effective section of the .080-inch 2024-T42 clad sheet, 2 inches wide with both edges fixed.

$$\frac{b}{t} = \frac{2.0}{.08 \times .96} = 26.1, \text{ based on 2 percent cladding per side of sheet}$$

$$F_{cc} = 39,000 \text{ p.s.i.} \qquad \text{(Reference 8, Figure 403:18)}$$

$$F_{cy} = 34,000 \text{ p.s.i.} \qquad \text{(Reference 7, Table 3.2.3.0(d))}$$

Loads on point A are critical.

$$M.S. = \frac{F_{cy}}{f_{bu} + f_c} - 1 = \frac{34,000}{(26,800 + 604)} - 1 = +0.24$$

CROTCH STRAP TUBE (Reference Drawing 201-00038)

The crotch strap is mounted on a steel tube, positioned below the seat pan, and attached to each side of the bucket. The crotch strap tube is also utilized to react down loads induced by the seat occupant. The critical design loads are from the crotch strap.

Crotch strap design load = 2,200 pounds (See Table IV)
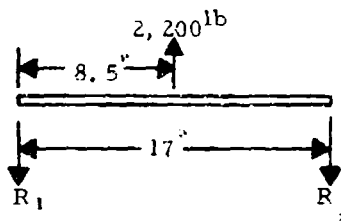
1-inch-diameter tube, t=0.058 in.
4340 stl., F_{tu} = 200,000 p.s.i.
A = 0.172 in.^{2}

$\frac{D}{t}$ = 17.21, I = 0.01911 in.^{4}

$$M_m = \frac{2,200}{2} \frac{(17)}{2} = 9,350 \text{ in.-lb.}$$

$$R_1 = R_2 = \frac{2,200}{2} = 1,100 \text{ lb.}$$

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$$f_{bu} = \frac{Mc}{I} = \frac{9350(0.50)}{0.01911} = 244,000 \text{ p.s.i.}$$

$$\frac{D}{t} = 17.21 \text{ and } F_{tu} = 200,000 \text{ p.s.i.}$$

$$F_{bu} = 259,000 \text{ p.s.i.} \qquad \text{(Reference 7, Figure 2.4.1.1.1)}$$

$$f_{su} = \frac{2R1}{A} = \frac{2(1,100)}{.172} = 12,790 \text{ p.s.i.}$$

$$F_{su} = 119,000 \text{ p.s.i.} \qquad \text{(Reference 7, Table 2.3.11(a))}$$

$$M.S. = \frac{1}{\sqrt{R_b^2 + R_s^2}} - 1$$

where

$$R_b = \frac{f_{bu}}{F_{bu}} = \frac{244,000}{259,000} = 0.942$$

$$R_s = \frac{f_{su}}{F_{su}} = \frac{12,790}{119,000} = 0.1074$$

$$M.S. = \frac{1}{\sqrt{0.885 + .115}} - 1 = \frac{1}{1} - 1 = 0$$

LAP BELT ATTACHMENT (Reference Drawing 201-00039)

The design load for the lap belt is 5000 pounds. The resultant design load on each fitting is 2500 pounds.

The fitting and mating structure are designed such that only the rear tang resists P_x.

When $\alpha = 45°$,

$$P_x = P_z = (\sin 45°) \, 2,500 = 1,770 \text{ lb.}$$

$$R_{z_2} (1.313) = 1,770 (1.23) - 1,770 (0.57)$$

$$R_{z_2} = 885 \text{ lb.}$$

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$P_{(Lap\ Belt)}$

P_z

0.428"

P_x

0.06"

θ

Reference page 75
for Lug Side View

0.10"

0.032"

0.04"
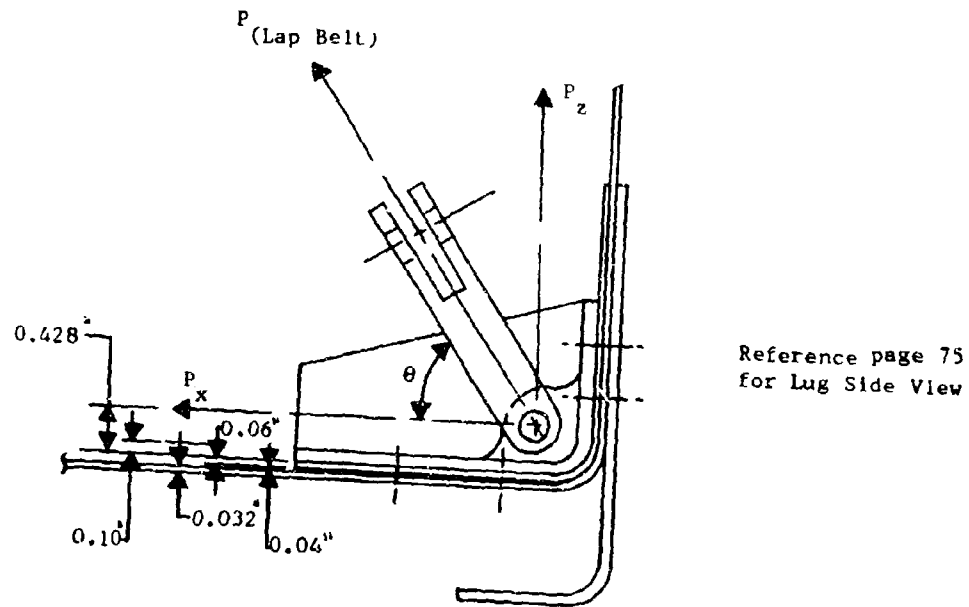
(Reference Drawings 201-00039, 201-00045)

A

E

E

B

B

B

B

B

A

C

C

D

C

D

C

C
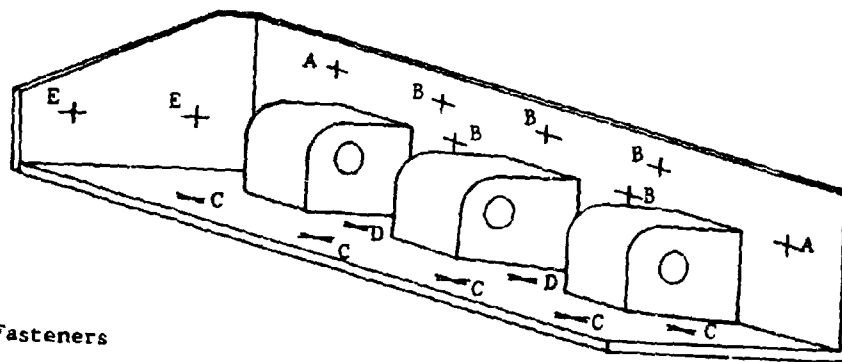
Fasteners

A - MS20470-DD6
B - MS20426-DD6
C - MS90354-3/16
D - MS90353-3/16
E - MS90354-5/32

Figure 23. Lap Belt Attachment.

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$$R_{z_1} = P_z - R_{z_2} = 885 \text{ lb.}$$

$$R_x = P_x = 1,770 \text{ lb.}$$

Similarly, for $\alpha = 55^\circ$,

$$P_x = (\cos 55^\circ)(2500) = 1,435 \text{ lb.}$$

$$P_z = (\sin 55^\circ)(2500) = 2,050 \text{ lb.}$$

$$R_{z_2} (1.313) = 1,435 (1.23) - 2,050 (0.57)$$

$$R_{z_2} = 456 \text{ lb.}$$

$$R_{z_1} = P_z - R_{z_2} = 1,594 \text{ lb.}$$

$$R_x = P_x = 1,435 \text{ lb.}$$

75

JOINT A

Shear-Bearing

$$P_u{}' = K_{br} A_{br} F_{tu} \qquad \text{(Reference 16)}$$

where

$$A_{br} = Dt = 0.3125(2)\ 0.141 = 0.0882\ \text{in.}^2$$

$$F_{tu} = 64,000\ \text{p.s.i.} \qquad \begin{array}{l}\text{(Reference 7,}\\ \text{Table 3.2.3.0(b))}\end{array}$$

To find K_{br},

$$\frac{a}{D} = \frac{0.438}{0.3125} = 1.4,\ K_{br} = 1.35$$

$$P_u{}^l = 1.35\ (0.0882)(64,000) = 7,610\ \text{lb.}$$

Assume a wear factor of 2 due to the steel male fitting and the aluminum female fitting.

$$M.S. = \frac{7.610}{2(2,500)} - 1 = +0.52$$

JOINT B

Pin Analysis, NAS 1588-3, 3/16-dia. bolt

$$F_{tu} = 185,000\ \text{p.s.i.} \quad P_{s_{su}}{}^l = 3,100\ \text{lb.}$$
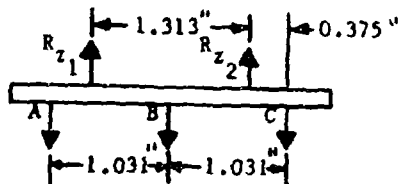
Assume that the beam is pinned at point B.

P_{max} occurs for $\alpha = 55°$, $R_{z_1} = 1,594$ lb.

$$P_A = \frac{(1.031-0.375)}{1.031}\ 1,594 = 1,015\ \text{lb.}$$

PIN SHEAR

$$P_s{}^l{}_{su} = 3,680\ \text{lb.} \qquad \begin{array}{l}\text{(Reference 21,}\\ \text{Table D1.1, AN-4 bolt)}\end{array}$$

$$M.S. = \frac{P^l_{s\,su}}{P_{ssu}} - 1 = \frac{3,080}{1,015} - 1 = 2.04$$

LUG SHEAR BEARING

$$P_u = K_{br} A_{br} F_{tu} \qquad \text{(Reference 16)}$$

where

$$A_{br} = Dt = 0.25(0.25) = 0.0625 \text{ in.}^2$$

$$F_{tu} = 64,000 \text{ p.s.i.}$$

To find K_{br},

$$\frac{a}{D} = \frac{0.25}{0.25} = 1.00$$

$$K_{br} = 0.75 \qquad \text{(Reference 16, Figure 11)}$$

$$P^l_u = 0.75 (0.0625) \, 64,000 = 3,000 \text{ lb.}$$

$$M.S. = \frac{P'_u}{P_u} - 1 = \frac{3,000}{1,015} - 1 = 1.96$$

Base Fitting 201-00045

Utilizing the fitting and associated side and bottom skins as a pinned end beam, the lap belt loads are beamed forward to the support hat (201-00042) and aft to the vertical posts.

Section A-A, see page 67
Assumed Effective Section

All Sheet
is 2024-T42 Clad

0.063" 0.032"
0.08"
0.175"
1.125"
0.135"
0.04"
2.25"
1.125"

$$\overline{y} = 1.10 \text{ in.}$$

$$I = .1964 \text{ in.}^4$$

1"

77
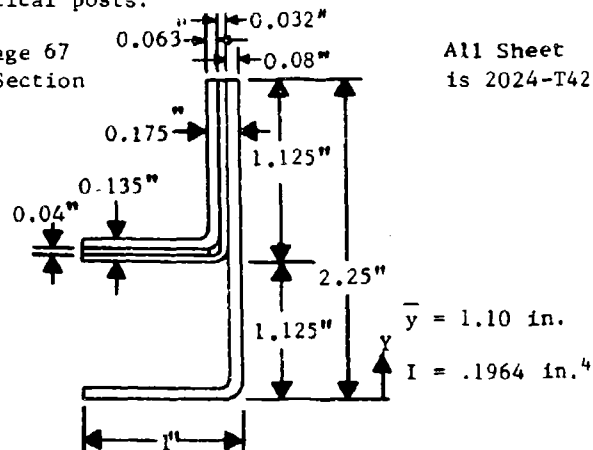
$$M_{A-A} = \frac{2,200}{2} (12.25-7.0) = 5,790 \text{ in.-lb.}$$

$$f_{bu} = \frac{Mc}{I} = \frac{5,790(2.25-1.10)}{0.1964} = 33,800 \text{ p.s.i.}$$

Compression in top element,

$$\frac{b}{t} = \frac{2.25-1.1}{0.175} = 6.56, \text{ cladding } t_c = 8\%(t)$$

$$\frac{b}{t_{eff}} = \frac{6.56}{(1-0.08)} = 7.15$$

$$F_{cc} = 45,000 \text{ p.s.i.} \qquad \text{(Reference 8, Figure 403:18)}$$

$$F_{c_y} = 34,000 \text{ p.s.i.} \qquad \text{(Reference 7, Table 3.2.3.0(d))}$$

$$M.S. = \frac{F_{cy}}{f_{bu}} -1 = \frac{34,000}{33,800} -1 = +0.01$$

The side skins (t = 0.036 in. - 2024-T42) are utilized to react by skin tension the loads tending to rotate the bucket lip down plus loads normal to the skins.
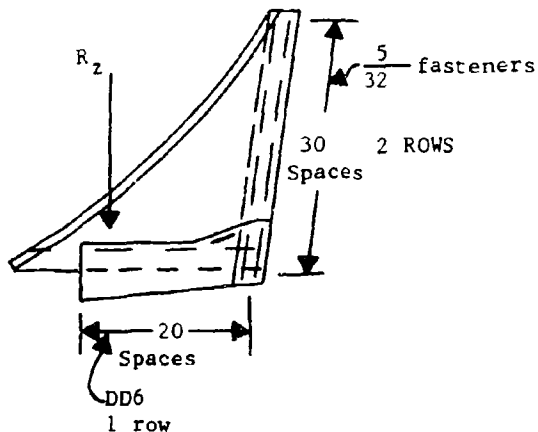
(see Drawing 201-00032)

Skin tension load due to down load
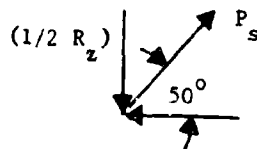
$R_z = 2,900$ lb.

(see page 54)

$$P_s = \frac{\frac{1}{2}R_z}{\sin 50°} = 1,895 \text{ lb.}$$

Skin normal load due to side load,

$$P_L = 4,000 \text{ lb.} \qquad \text{(see page 55)}$$

Conservatively assume that one-half of one row of fasteners at the back posts and one-half of the single row at the bottom react the side load of 4,000 lb.; also consider approximately 7 inches of skin reacting $P_s = 1,895$ lb.

Back Posts

29-AD5 fasteners, 30 spaces

$$P^1_{bru} = 450 \text{ lb. in 2024-T42} \qquad \text{(Reference 8,}$$
Tables 702:1101,
702:1402)

$$P^1_{tu} = 311 \text{ lb.}$$

Seat Bottom

21-DD6 fasteners, 22 spaces

$$P^1_{bru} = 525 \text{ lb.}$$

$$P^1_{tu} = 354 \text{ lb.}$$

$$P_{bru} = \frac{1.895}{\frac{7}{0.75}} = 203 \text{ lb. per rivet, 0.75-in. spacing}$$

$$P_{tu} = \frac{4000}{(21+29)\frac{1}{2}} = 160 \text{ lb. per rivet}$$

AD5 fasteners

$$M.S. = \frac{1}{R_s + R_t} - 1$$

$$R_s = R_{bru} = \frac{P_{bru}}{P^1_{bru}} = \frac{203}{450} = 0.451, \quad R_t = \frac{P_{tu}}{P^1_{tu}} = \frac{160}{311} = 0.515$$

$$M.S. = \frac{1}{0.966} - 1 = +0.05$$

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SEAT BUCKET ATTACHMENT AND SLIDE ASSEMBLY

The seat bucket is attached to the support structure by two combined lateral and vertical slide assemblies. The slide assemblies are critical when positioned at the top adjust position for Condition 4, Combined Loads.

LOADS (see Figure 20, page 66 and Figure 24, page 81)

$$R_{UL} = 2520 \text{ lb.}$$

$$R_{UR} = 3430 \text{ lb.}$$

$$R_{YU} = 688 \text{ lb.}$$

$$R_{LL} = -1208 \text{ lb.}$$

$$R_{LR} = -2652 \text{ lb.}$$

$$R_{YL} = 312 \text{ lb.}$$

Friction loads, F, are based on a friction coefficient μ of 0.10.

Example:

$$F_{LR} = \mu |R_{LR} + R_{YL}|$$

$$F_{LR} = 296 \text{ lb.}$$

$$F_{LL} = 152 \text{ lb.}$$

$$F_{UR} = 412 \text{ lb.}$$

$$F_{UL} = 321 \text{ lb.}$$

The load diagram, Figure 24, is derived by transmitting the z' components of occupant, seat, and armor loads to the seat back. The load then passes through the lower slide fitting, 201-00029, through the actuator, Load A_2, and to the support structure via the attenuators, Load A. The remaining components are reacted in a normal manner. Friction load, F, is applied at each sliding joint.

Figure 24. Seat Bucket Attachment Load Diagram.
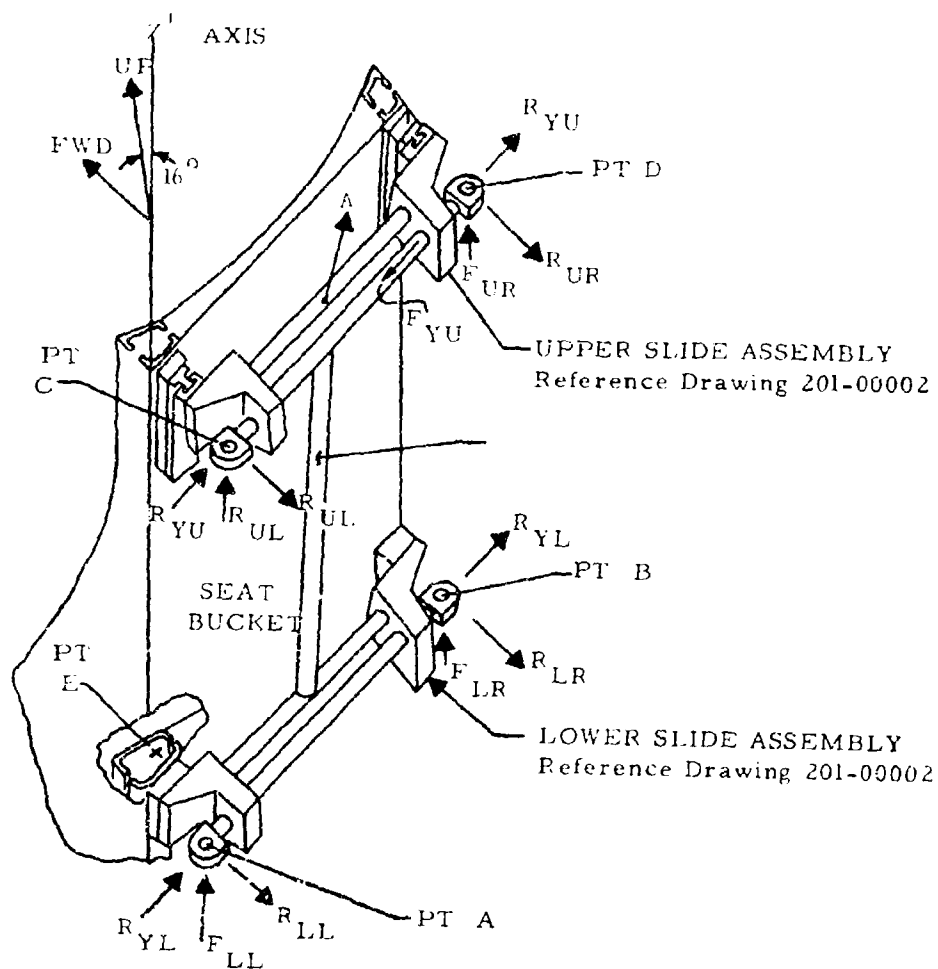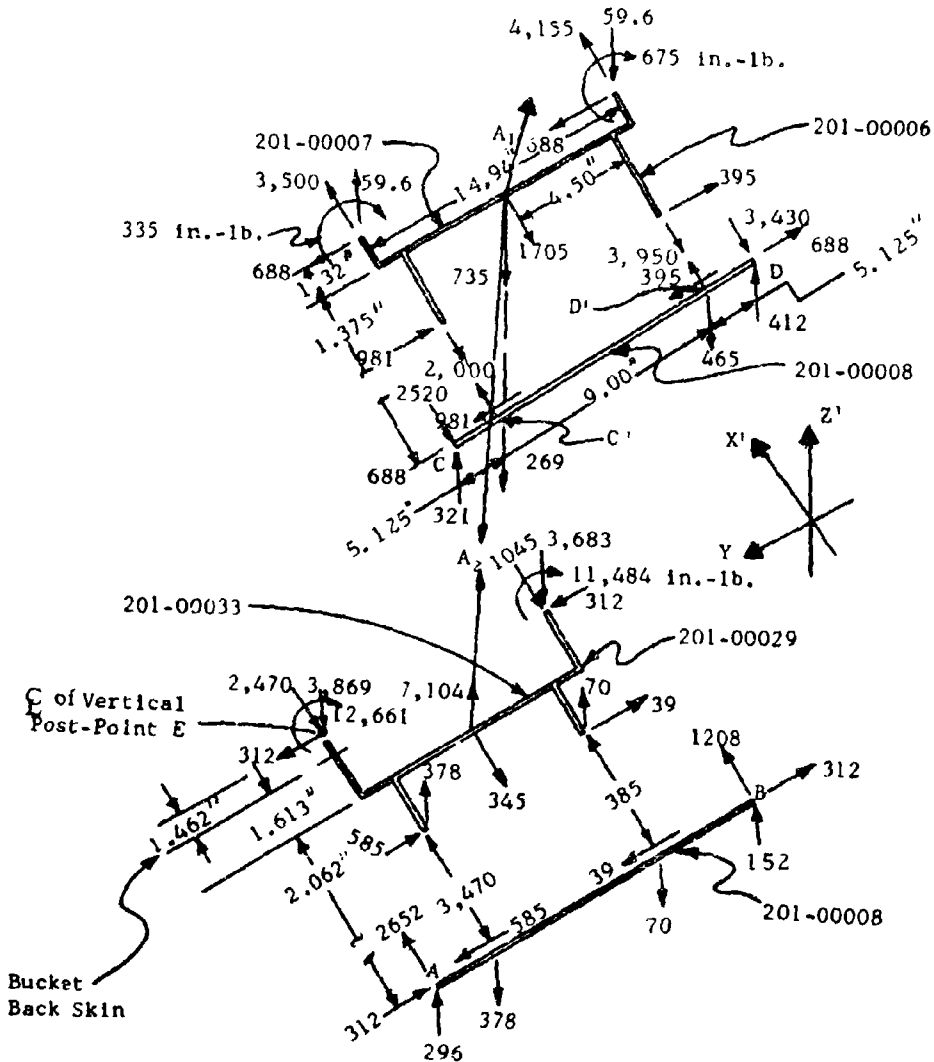
Figure 25. Schematic Diagram of Bucket Attachment Loads, Condition 4.

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LATERAL SLIDE FITTING (Reference Drawing 201-00006)
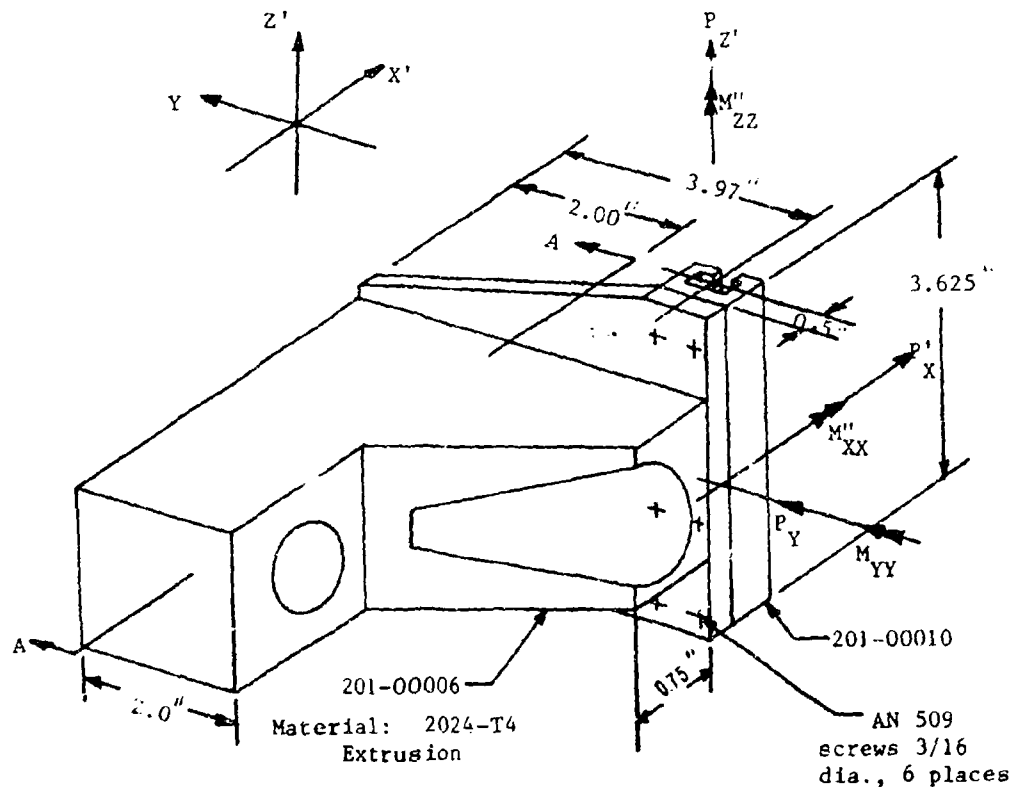
Figure 26. Upper Lateral Slide Fitting.

LOADS, Condition 4 (see Table VII)

$P'_X = R'_{UR} = 4,155$ lb.

$M'_x = 0$ Right-Hand Rules Applies

$P'_y = 688$ lb. $P'_z = -59.6$ lb.

$M_Y = -M'_Y = -675$ in.-lb. $M'_z = 0$

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SECTION A-A

$A = 1.17 \text{ in}^2$

$\bar{Z}' = 0.823 \text{ in.}$

$I_Z' = 0.190 \text{ in}^4$

$I_X' = 0.505 \text{ in}^4$
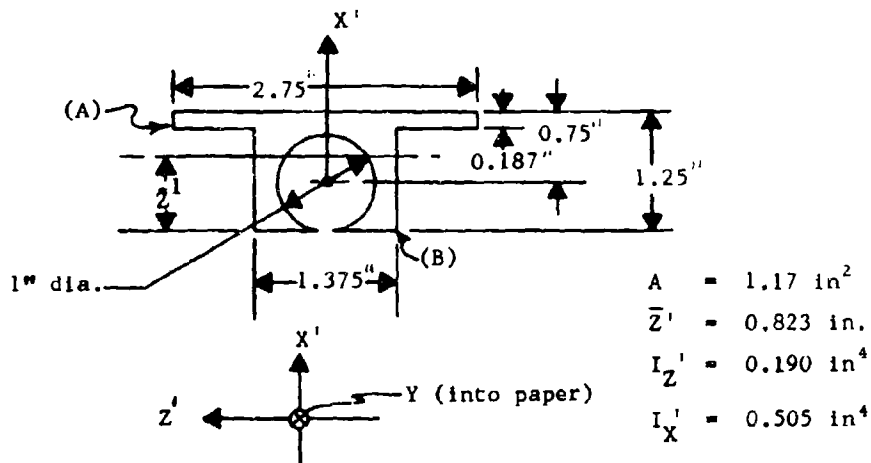
$P_x' = 4185 \text{ lb.}$

$M_x' = -(-59.6)\,1.97 = 117.5 \text{ in.-lb.}$

$P_y = 688 \text{ lb.}$

$M_y = -675 - (-59.6)(0.5 + 1.25 - 0.823) = -620 \text{ in.-lb.}$

$P_z' = -59.6 \text{ lb.}$

$M_z' = 4{,}155\,(1.97) + 688\,(0.5 + 1.25 - 0.823) = 8{,}838 \text{ in.-lb.}$

The maximum compressive stress occurs at Point (A).

$$f_{bu} \atop (A) = -(M_z')\,\frac{(1.25 - 0.823)}{I_{zz}} + M_x'\,\frac{(1.375)}{I_X}$$

$$= \frac{-8{,}838}{0.190}(0.427) - 117.5\,\frac{(1.375)}{0.505} = -20{,}120 \text{ p.s.i.}$$

The maximum tensile stress occurs at Point (B).

$$f_{bu} \atop (B) = 8{,}823\,\frac{(0.823)}{0.190} - 117.5\,\frac{(0.6875)}{0.505} = 38{,}040 \text{ p.s.i.}$$

$$f_c = -\frac{P_y}{A} = -\frac{688}{1.19} = -577 \text{ p.s.i.}$$

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$$f_s = \frac{P_x' + P_z'}{A} = \frac{4155 + 59.6}{1.19} = 3,500 \text{ p.s.i.}$$

Assume that the torque, $M_y = -620$ in.-lb., is transmitted by differential bending.

$$M_c = \frac{620\ (1.97)}{1.375/2} = 1,775 \text{ in.-lb.}$$

$$f_{bu_{(B)}} = \frac{1775\ (0.823)}{0.190/2} = +15,350 \text{ p.s.i.}$$

$$\Sigma f_{bu_{(B)}} = 38,040 + 15,350 = 53,390 \text{ p.s.i.}$$

$$F_{tu} = 57,000 \text{ p.s.i.} \hspace{3cm} \text{(Reference 7)}$$

$$F_{su} = 39,000 \text{ p.s.i.} \hspace{3cm} \text{(Reference 7)}$$

$$M.S. = \frac{1}{\sqrt{R_b^2 + R_s^2}} - 1$$

where

$$R_b = \frac{53,390 - 577}{57,000} = 0.93$$

$$R_s = \frac{3,500}{F_{su}}$$

$$R_s = \frac{3,500}{39,000} = 0.09$$

$$M.S. = \frac{1}{0.934} - 1 = +0.07$$

UPPER SLIDE FITTING SUPPORT TUBE (Reference Drawing 201-00007)

The upper slide fitting support tube is designed to support the slide fittings, 201-00006, for the lateral load condition. As an overlapping assumption, it is assumed for the remaining load conditions that the support tube is cut inboard of the slide fittings (reference Figure 28). This is a conservative assumption for the lateral slide tube, 201-00008. However, preliminary analysis has shown that as the slide assemblies are loaded and the structure deflects, the lateral slide tube relieves the fitting support tube of load in the X-Y plane. This is due to the slide fittings' binding against the lateral slide tubes at position C' and D' (reference Figure 25).

For Condition 3, lateral loading, the following loads are obtained in the

same manner as in Figure 24.
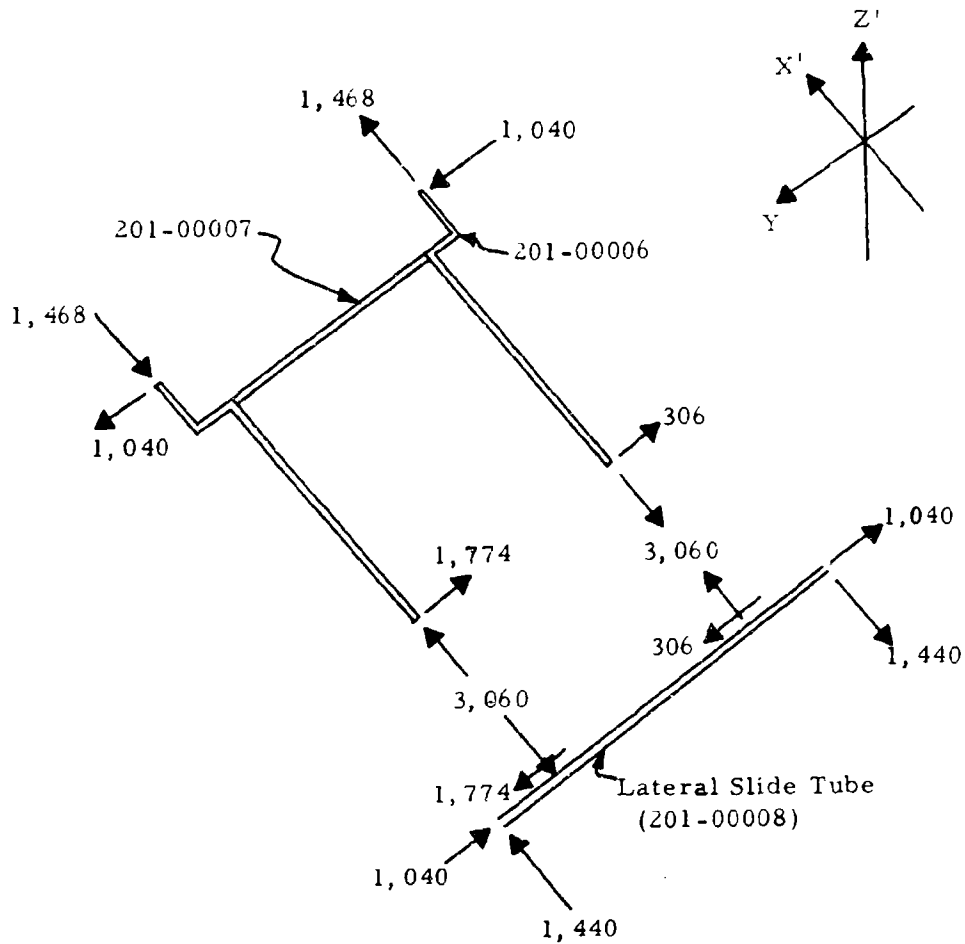
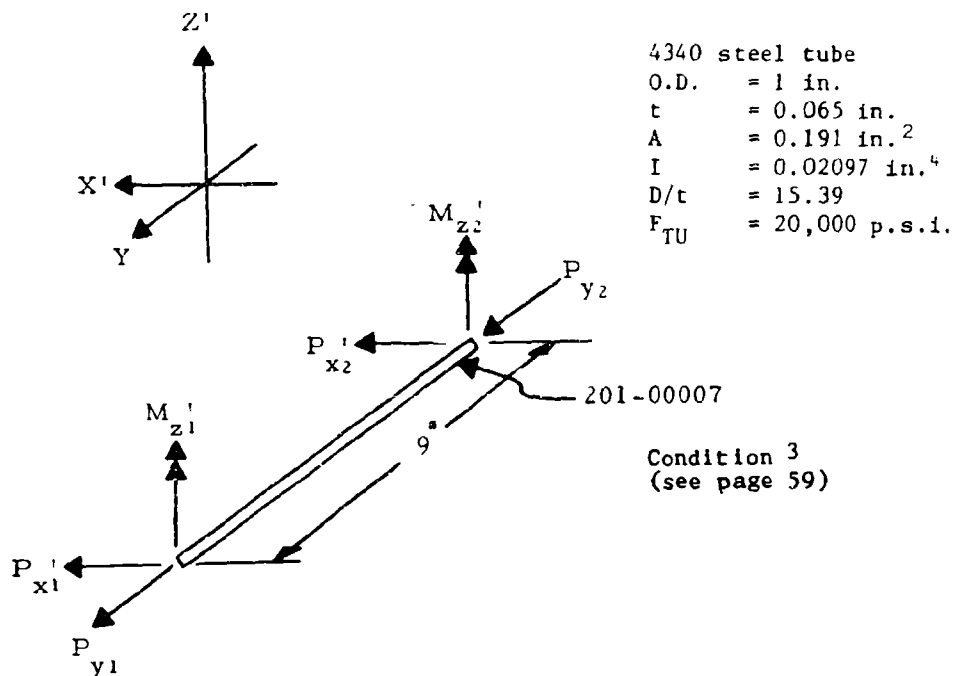Figure 27. Schematic Diagram of Bucket Attachment Loads, Condition 3.

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UPPER SLIDE FITTING SUPPORT TUBE (Reference Drawing 201-00007)

4340 steel tube
O.D. = 1 in.
t = 0.065 in.
A = 0.191 in.^{2}
I = 0.02097 in.^{4}
D/t = 15.39
F_{TU} = 20,000 p.s.i.

201-00007

Condition 3
(see page 59)

$$M_{max} = M'_{Z_1} = -8,185 \text{ in.-lb.}$$

$$f_{bu} = \frac{Mc}{I} = \frac{8,185(0.5)}{0.02097} = 203,500 \text{ p.s.i.}$$

$$\frac{D}{t} = 15.39, \quad F_{bu} = 263,000 \text{ p.s.i.} \qquad \text{(Reference 7, Figure 2.4.1.1.1)}$$

$$f_c = \frac{P}{A} = \frac{734}{0.191} = 3,840 \text{ p.s.i.}$$

The shear load is small; therefore,

$$M.S. = \frac{F_{bu}}{f_{bu} + f_c} - 1$$

$$= \frac{263,000}{207,340} - 1 = +0.27$$

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LATERAL SLIDE TUBE (Reference Drawing 201-00008)

3, 500 lb   4, 155 lb

688 lb ← ... 2.97"

1.705/2   1.705/2   ← 688 lb

2.695"   cut

688 →   ← 688

M₁   M₂

2,648 lb   3,303 lb

X' - Y Plane

4340 steel tube
O.D. = 1.375 in.
t   = 0.083 in.
A   = 0.3369 in.²
I   = 0.0706 in.⁴
D/t   = 16.58
F_{TU} = 200,000 p.s.i.

M₁ M₂   (A)

201-00008

R₁   19.25"   R₂
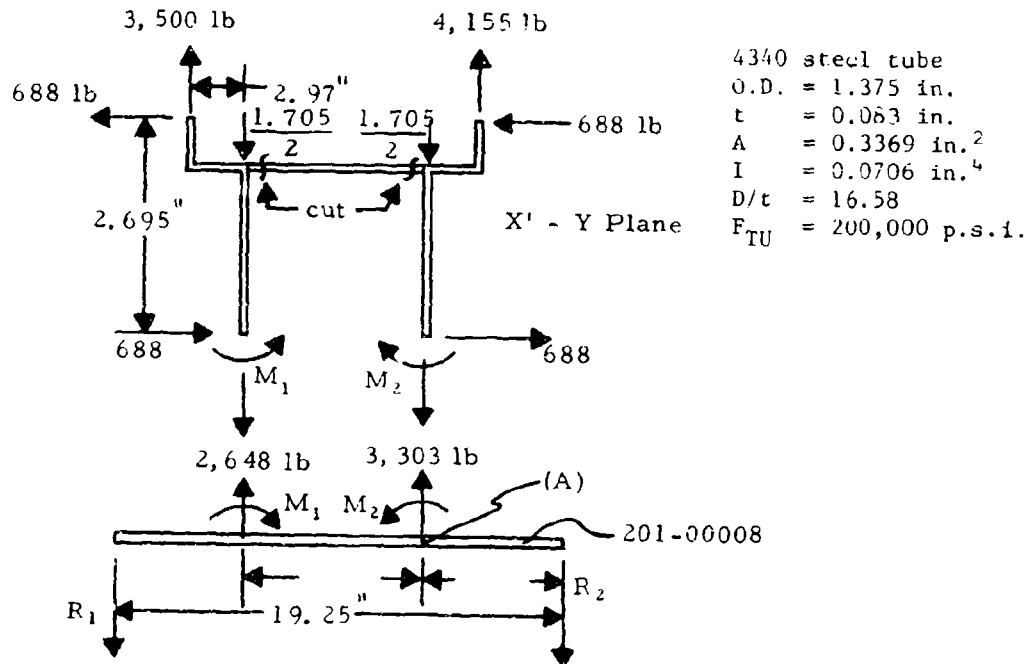
Figure 28.   Upper Slide Load Schematic, Condition 4.

M_1 = 3500(2.97) -688(2.695) = 8,550 in.-lb.

M_2 = 4155(2.97) +688(2.695) = 14,200 in.-lb.

R_1 = $\left[-14200 +8550 +3303(5.125) +2648(14.125)\right] \dfrac{1}{19.25}$

  = $\dfrac{48650}{19.25}$ = 2520 lb.(ck)

5.125   269 lb   465 lb
        9
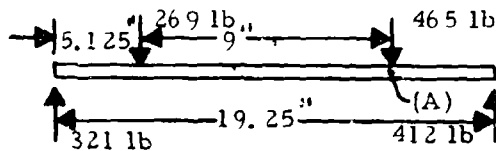
Y - Z' plane
(see Figure 27)

321 lb   19.25"   412 lb   (A)

$M_{max(A)}$ = 412(5.125) = 2,110 in.-lb.

ΣM = 17,600 + 2,110 = 17,140 in.-lb.

$\dfrac{D}{t}$ = 16.58, F_{bu} = 260,000 p.s.i.   (Reference 7, Figure 2.4.1.1.1)

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$$f_{bu} = \frac{Mc}{I} = \frac{17,740(1.375)}{2(0.0706)} = 172,500 \text{ p.s.i.}$$

$$f_c = \frac{P}{A} = \frac{2(688)}{0.3369} = 4,090 \text{ p.s.i.}$$

$$f_s = \frac{2P}{A} = \frac{2(3430 + 412)}{0.3369} = \frac{6910}{0.3369} = 20,520 \text{ p.s.i.}$$

$$F_{su} = 119,000 \text{ p.s.i.}$$

(Reference 7, Table 2.3.1.1(a))

$$M.S. = \frac{1}{\sqrt{R_b^2 + R_s^2}} - 1$$

where

$$R_b = \frac{172,500 + 4,090}{260,000} = 0.679$$

$$R_s = \frac{20,520}{119,000} = 0.173$$

$$M.S. = \frac{1}{0.7} - 1 = +0.43$$

LUG ANALYSIS (Reference Drawing 201-00006)
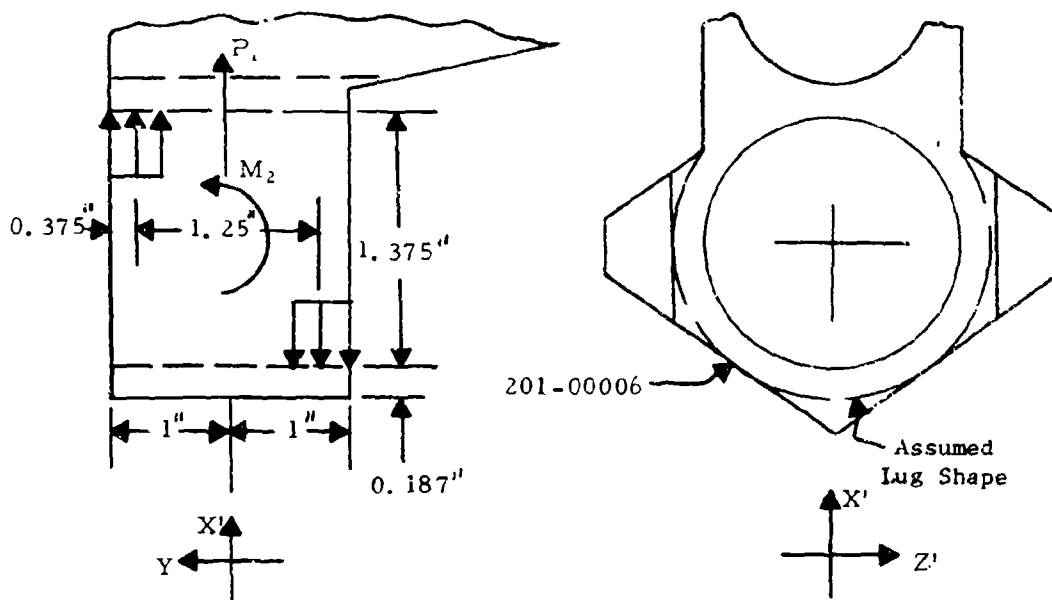
Figure 29. Lug, Upper Slide Fitting.

$$P_1 = 3,303 \text{ lb.}$$

$$M_2 = 14,200 \text{ in.-lb.}$$

$$P_u = \frac{M_2}{1.25} + 3,303 = 11,350 + 3,303 = 14,653 \text{ lb.}$$

$$P'_u = K_{br} A_{br} \, F_{TU}$$

where

$$F_{tu} = 57,000 \text{ p.s.i.}$$

$$A_{br} = Dt = 1.375(2) \, 0.375 = 1.031 \text{ in.}^2$$

$$K_{br} \quad \frac{a}{D} = \frac{1.375/2 + 0.187}{1.375} = 0.635$$

$$K_{br} = 0.25 \qquad \text{(Reference 16, Figure 11)}$$

$$P'_u = 0.25(1.031) \, 57,000 = 14,700 \text{ lb.}$$

$$M.S. = \frac{P'_u}{P_u} - 1 = \frac{14,700}{14,653} - 1 = +0.01$$

SEAT ADJUSTMENT ACTUATOR (Reference Drawing 201-00002)

The seat adjustment actuator is a Linear Electro-Mech Actuator (drawing number D-1913) of the Electronic Specialty Company, Los Angeles, California. It has an ultimate design load of 8,000 pounds tension or compression over a stroke of 5 inches.

$$P_{ACTUATOR} = A_2 \qquad \text{(see Figure 25)}$$

$$\text{Assume } \quad P_{ACTUATOR} = \frac{7553}{\sin 3/.22^\circ} = 7,580 \text{ lb.}$$

$$M.S. = \frac{8,000}{7,580} - 1 = +0.05$$

LOWER SLIDE ASSEMBLY ATTACHMENT TO SEAT BACK (Reference Drawing 201-00032)
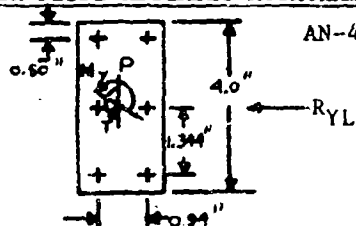
AN-4 Bolt (6)

To determine bolt loading (1.462 in aft of Vertical Post ₵), the following loads are determined from Condition 4.

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$$M_y = 7,011 \text{ in.-lb.}$$

$$T = 14,15? \text{ in.-lb.} \quad \text{(considering the beam} - \\ 201\text{-}00029 \text{ as a fixed} \\ \text{end beam)}$$

$$R_{YL} = 312 \text{ lb.}$$
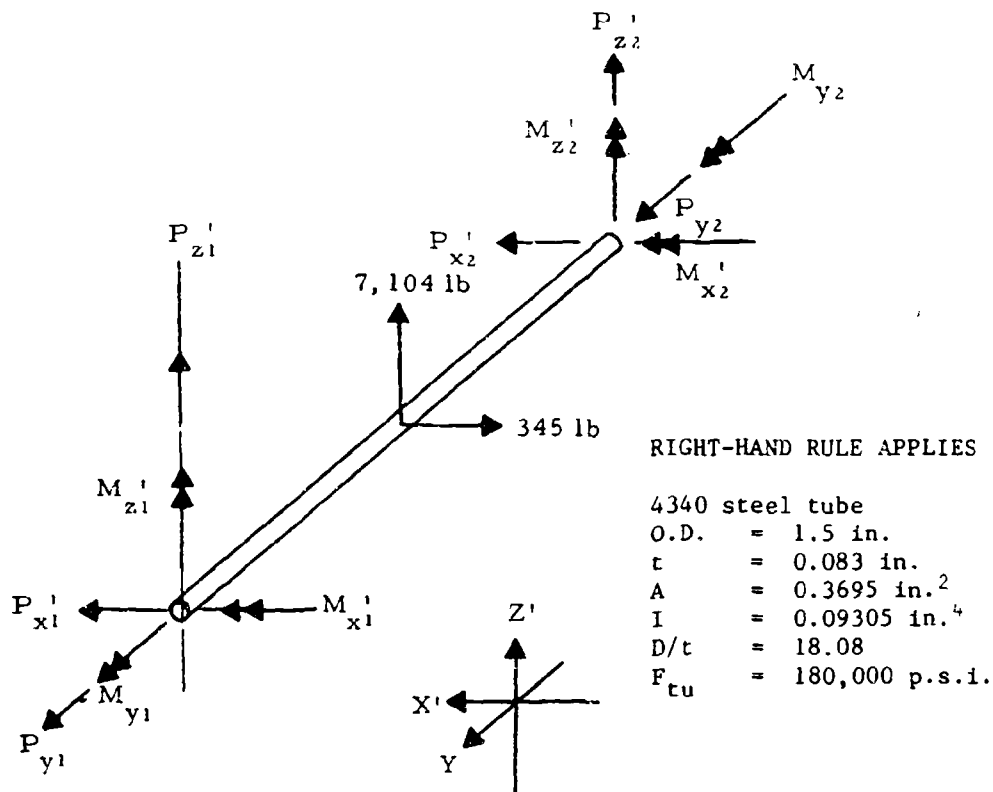
$$P = 3,869 \text{ lb.}$$

Using standard design procedures, the critical combined bolt load is $P_{tu} = 830$ lb. and $P_{su} = 2,930$ lb.

For the AN-4 bolt with a shear load $P_{su} = 830$ lb., the allowable tension load $P_{tu} = 3,500$ lb. (Reference 7, Figure 8.1.1.1.1).

$$M.S. = \frac{P_{tu}'}{P_{tu}} - 1 = \frac{3,500}{2,930} - 1 = +0.12$$

LOWER SLIDE FITTING SUPPORT TUBE (Reference Drawing 201-00033)

RIGHT-HAND RULE APPLIES

4340 steel tube

O.D.	=	1.5 in.
t	=	0.083 in.
A	=	0.3695 in.^2
I	=	0.09305 in.^4
D/t	=	18.08
F_{tu}	=	180,000 p.s.i.

Reference page 82 for loads and dimensions.

$P_{x_1}' = 3,470 - 2,470 = 1,000$ lb.

$P_{y_1} = -585 + 312 = -273$ lb.

$P_{z_1}' = 378 - 3,869 = -3,491$ lb.

$P_{x_2}' = 385 - 1,045 = -660$ lb.

$P_{y_2} = 39 - 312 = -273$ lb.

$P_{z_2}' = 70 - 3,683 = -3,613$ lb.

$M_{x_1}' = -3,869(2.97) = -11,490$ in.-lb.

$M_{y_1} = 3,869(3.075) - 12,661 + 378(2.062) = 0$ in.-lb.

$M_{z_1}' = 2,470(2.97) + 312(3.075) + 585(2.062) = +9,495$ in.-lb.

$M_{x_2}' = 3,683(2.97) = 10,950$ in.-lb.

$M_{y_2} = 3,683(3.075) - 11,484 + 70(2.062) = 0$ in.-lb.

$M_{z_2}' = -1,045(2.97) + 312(3.075) + 39(2.062) = -2,059$ in.-lb.

X' - Y plane

9, 495   2, 059
272   272
4. 5"   4. 5"
1, 000 lb   345   660

$M_{Max} = 9,495$ in.-lb.

Y - Z' plane

7, 104
11, 490   10, 950
3, 091   3, 613
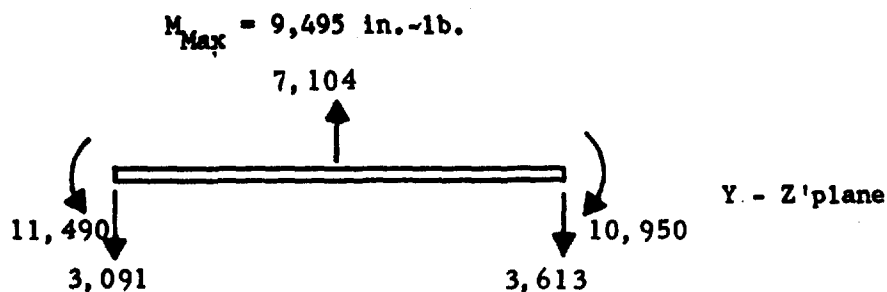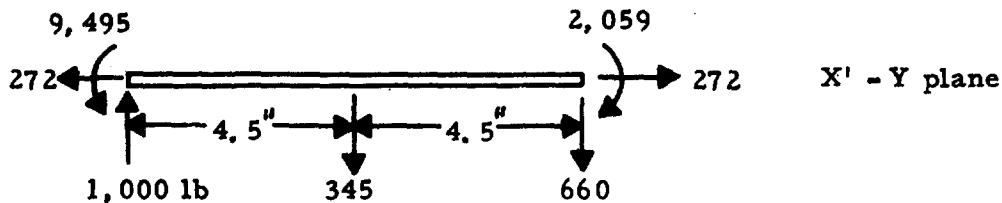
$M_{Max} = M_{\mathcal{C}} = 11,490 + 3,491(4.5) = 27,200$ in.-lb.

$M_{Total} = 5,029 + 27,200 = 27,700$ in.-lb.

$f_{bu} = \dfrac{Mc}{I} = \dfrac{27,700(0.75)}{0.09305} = 223,000$ p.s.i.

For $\dfrac{D}{t} = 18.08$, $F_{tu} = 180,000$ p.s.i., $F_{bu} = 230,000$ p.s.i.

(Reference 7, Figure 2.4.1.1.1)

$f_{su} = \dfrac{2P}{A} = \dfrac{2(3869 + 645)}{0.3695}$

$= \dfrac{2}{0.3695}(3,920) = 21,300$ p.s.i.

$F_{su} = 109,000$ p.s.i.

(Reference 7, Table 2.3.1.1(a))

$M.S. = \dfrac{1}{\sqrt{R_b^2 + R_s^2}} - 1$

where

$R_b = \dfrac{f_{bu}}{F_{bu}} = \dfrac{223,000}{230,000} = 0.969$

$R_s = \dfrac{f_{su}}{F_{su}} = \dfrac{21,300}{109,000} = 0.195$

$M.S. = \dfrac{1}{\sqrt{0.976}} - 1 = +0.01$

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Forward Support Posts (Reference Drawing 201-00003)
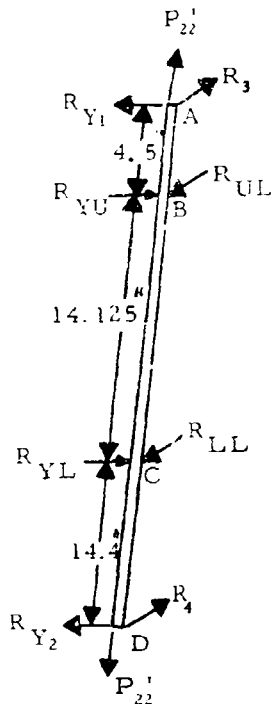
Condition 4 is the critical
design condition (see Table
VII)

R_{UL} = +2,520 lb.

R_{LL} = -2,652 lb.

R_{YU} = 688 lb.

R_{YL} = 312 lb.

R_3 = 1,015 lb.

R_4 = -1,142 lb.

R_{Y1} = 730 lb.

R_{Y2} = 269 lb.

P_{22}' = 9,131 lb.

For the analysis, the loading is assumed to be as follows:
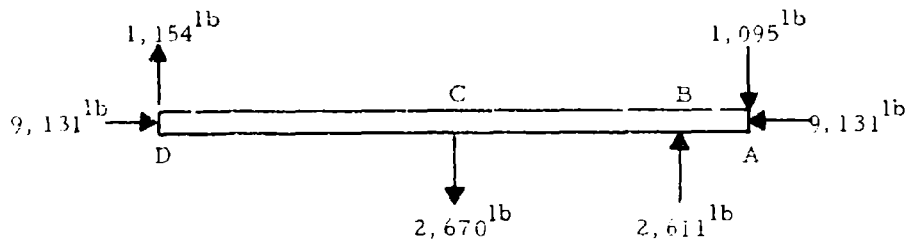
4130 Steel Tube	I = .0256 in.^{4}
O.D. = 2¼ in.	D/t = 45.9
t = .049 in.	I/y = .1825 in.^{3}
A = .3388 in.^{2}	F_{tu} = 180,000 p.s.i.

94

$$j = \sqrt{EI/P} = \sqrt{\frac{30(10^6)(.2056)}{9131}} = 26.1$$

$$L/j = \frac{33.025}{26.1} = 1.27$$

For 2670^{lb} Load

$$a = 14.4$$

$$b = 18.625$$

$$C_1 = \frac{Wj \sin b/j}{\sin L/j} = -\frac{(2670)(26.1)(.65560)}{0.95510} = -47,800$$

$$C_2 = 0$$

For 2611^{lb} Load

$$a = 28.525$$

$$b = 4.5$$

$$C_1 = -\frac{(-2611)(26.1)(.16918)}{0.95510} = +12,100$$

$$M_{dc} = (-47,800 + 12,100)(\sin x/26.1)$$

$$M_c = (-35,700)(.52269) = -18,700$$

$$M.S. = \frac{1}{R_c + \sqrt{R_b^2 + R_s^2}} - 1$$

$$f_{bu} = -\frac{M_{max}}{I/c} = -\frac{(-18,700)}{.1825} = 102,500 \text{ p.s.i.}$$

$$F_{bu} = 198,000 \text{ p.s.i.} \qquad \text{(Reference 7)}$$

$$R_b = \frac{f_{bu}}{F_{bu}} = \frac{102,500}{198,000} = 0.518$$

$$f_c = P/A = \frac{9,131}{.3388} = 26,900$$

$$\frac{L}{\rho} = \frac{33.0}{0.7782} = 42.4$$

$$F_c = 127,000 \text{ p.s.i.} \qquad \text{(Reference 7, Figure 2.4.2.3(c))}$$

95

$$R_c = \frac{f_c}{F_c} = \frac{26,900}{127,000} = 0.212$$

$$f_{su} = \frac{2P}{A} = \frac{2(1154)}{.3388} = 6,830 \text{ p.s.i.}$$
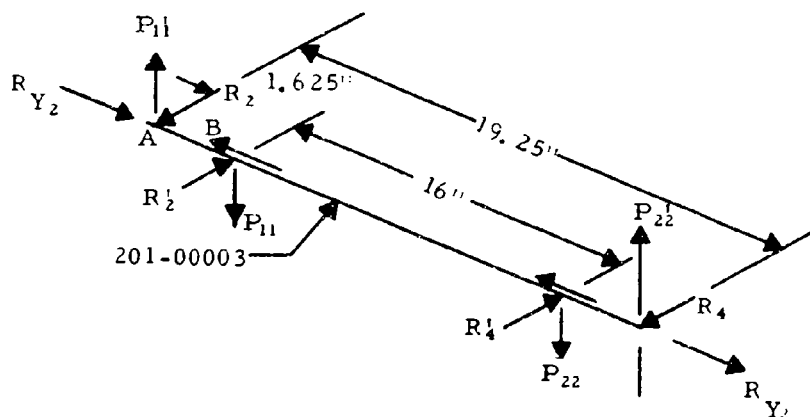
$$F_s = 109,000 \text{ p.s.i.}$$ (Reference 7, Table 2.3.1.1(a))

$$R_s = \frac{f_{su}}{F_{su}} = \frac{6,830}{109,000} = 0.063$$

$$M.S. = \frac{1}{0.212 + \sqrt{0.518)^2 + (0.063)^2}} - 1 = +0.37$$

Forward Crossover Tube (Reference Drawing 201-00003)

Loads, Condition 4, Combined Load (Reference Table VII)

$$P_{11}' = -12,875^{lb.} \qquad\qquad R_4' = -1,238^{lb.}$$

$$R_2 = -214^{lb.} \qquad\qquad P_{22} = -8,750^{lb.}$$

$$R_2' = -119^{lb.} \qquad\qquad R_{YU} = 688^{lb.}$$

$$P_{11} = -13,259^{lb.} \qquad\qquad R_{YL} = 312^{lb.}$$

$$P_{22}' = -9,131^{lb.} \qquad\qquad R_{Y_2} = 269^{lb.}$$

$$R_4 = -1,142^{lb.}$$

Material: 4130 Steel Tube

$$O.D. = 2\tfrac{1}{4}\ in. \qquad\qquad D/t = 45.9$$

$$t = 0.049\ in. \qquad\qquad I/y = 0.1825\ in.^3$$

$$A = 0.3388\ in.^2 \qquad\qquad F_{tu} = 180,000\ p.s.i.$$

$$I = 0.2056\ in.^4$$

It is assumed that the forward support posts are fixed at the forward cross-over tube and pinned at the top (reference Figure 20). Therefore, the lateral loads on the forward posts induce a moment into the crossover tube, M_A.

$$M_A = \tfrac{1}{2}R_{YL}\left[\frac{(14.4)^3 + 2(14.4)(33.025)^2 - 3(14.4)^2(33.025)}{(33.025)^2}\right]$$

$$+ \tfrac{1}{2}R_{YU}\left[\frac{(28.525)^3 + 2(28.525)(23.025)^2 - 3(28.525)^2(33.025)}{(33.025)^2}\right]$$

(Reference 9, page 109)

$$M_A = 3,505\ in.-lb.$$

97
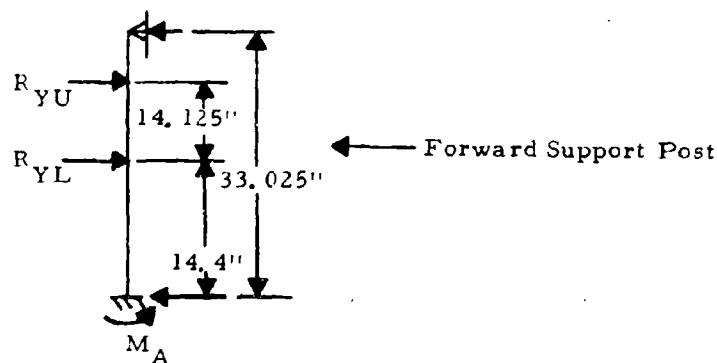
Due to the combined loading, the maximum moment in the forward crossover tube occurs at point B.

$$M_B = (12,875 + 214) \, 1.625 + 3,505 = 24,500 \text{ in.-lb.}$$

$$f_{bu} = \frac{Mc}{I} = \frac{24,500}{.1825} = 134,000 \text{ p.s.i.}$$

For

$$D_t = 45.9 \text{ and } F_{tu} = 180,000 \text{ p.s.i.}$$

$$F_{bj} = 198,000 \text{ p.s.i.} \qquad \qquad \text{(Reference 7, Figure 2.4.1.1.1)}$$

$$R_b = \frac{f_{bu}}{F_{bu}} = \frac{134,000}{198,000} = 0.678$$

$$f_{su} = \frac{2P}{A} = \frac{2(12,900)}{0.3388} = 75,000 \text{ p.s.i.}$$

$$F_{su} = 109,000 \text{ p.s.i.} \qquad \qquad \text{(Reference 7, Table 2.3.1.1(a))}$$

$$R_s = \frac{f_{su}}{F_{su}} = \frac{75,000}{109,000} = 0.688$$

$$M.S. = \frac{1}{\sqrt{R_b^2 + R_s^2}} - 1 = \frac{1}{\sqrt{(0.678)^2 + (0.688)^2}} - 1 = +0.04$$

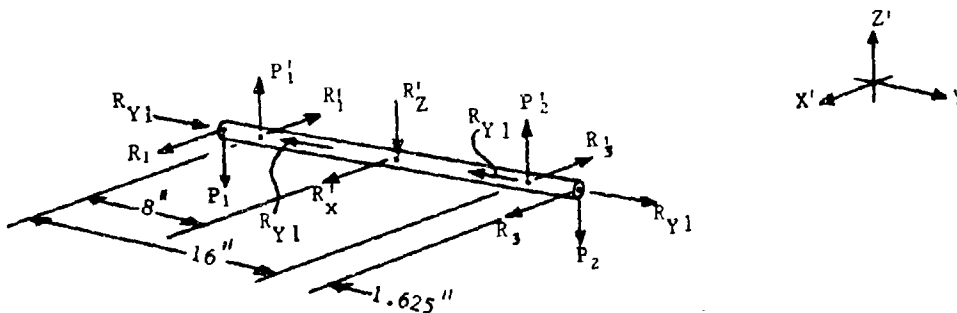UPPER CROSSOVER TUBE (Reference Drawing 201-00013)

Figure 30. Upper Crossover Tube.

98

R_1 = 2,434 lb.

P_1 = -12,201 lb.

R_{Y_1} = 730 lb.

P_1' = -9,390 lb.

R_1' = 3,601 lb.

R_x' = 2,050 lb.

R_z' = 6,370 lb.

P_2' = -4,960 lb.

R_3' = 1,906 lb.

R_3 = 1,015 lb.

P_2 = -8,514 lb.

9,390 lb.   6,370 lb.   4,960 lb.

8"

1.625"

Y - Z' plane

12,201 lb.

8,514 lb.
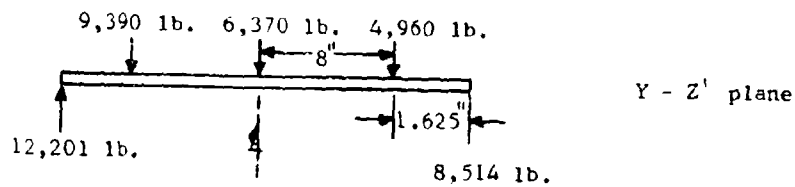
M_{max} (At $\mathcal{L}$) = 12,201 $(\frac{19.25}{2})$ - 9,390(8) = 42,380 in.-lb.

3,601 lb   2,050 lb   1,906 lb

8"   1.625"

X' - Y Plane

2,434 lb

1,015 lb
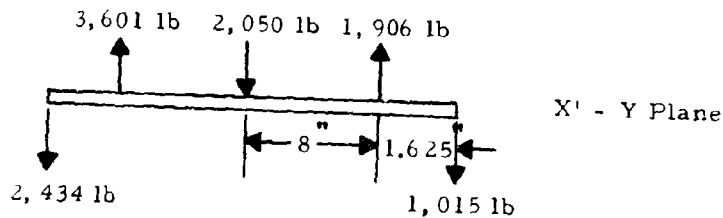
M_{max} (At $\mathcal{L}$) = 2,434 $(\frac{19.25}{2})$ - 3,601 (8) = -5,408 in.-lb.

Z''

Z'

17.8°

t = 0.065"

X''

0.75"

X'

O.D. = 1.75 in.

0.187 in.
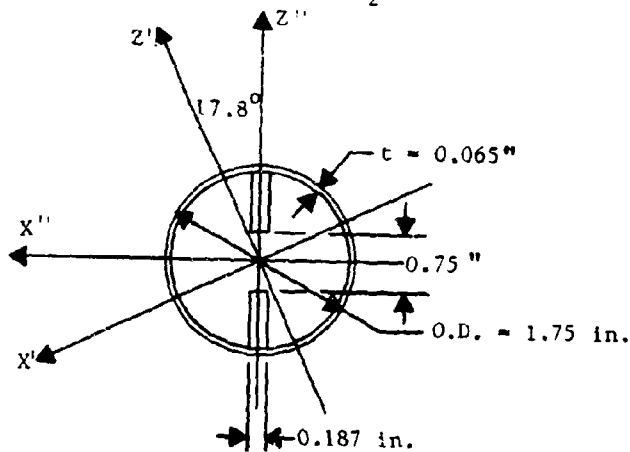
A = 0.488 in.^{2}

I_X'' = 0.178 in.^{4}

I_2'' = 0.122 in.^{4}

D/t = 26.95

F_{TU} = 200,000 p.s.i.

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$$M_x' = 42,380 \text{ in.-lb.}$$

$$M_t' = 5,408 \text{ in.-lb.}$$

$$M_{max} = M_x' + M_z' = 42,650 \text{ in.-lb.}$$

$$\text{Tan } \theta = \frac{5,408}{42,380} = 0.128, \quad \theta = 7.3^\circ$$

For $\phi = 17.8^\circ - 7.3^\circ = 10.5^\circ$

$$I = 0.178 (\cos^2\phi) + 0.122(\sin^2\phi)$$

$$= 0.178 (0.966) + 0.122(0.0331) = 0.172 + 0.00404 = 0.176 \text{in.}^4$$

$$f_{bu} = \frac{Mc}{I} = \frac{42,650(0.875)}{0.176} = 212,000 \text{ p.s.i.}$$

$$f_{su} = \frac{2P}{A} = \frac{2\left[(8,514 - 4,960) + (1,906 - 1,015)\right]}{0.488} \frac{(2)\ 3,660}{0.488} = 15,000 \text{ p.s.i.}$$

For $\frac{D}{t} = 26.95$ and $F_{tu} = 200,000$ p.s.i., $F_{bu} = 240,000$ p.s.i.

(Reference 7,
Figure 2.4.1.1.1)

For a rectangular section,

$$F_{bu} = F_{tu} + (K-1)\ \delta$$

where

$$F_{tu} = 200,000 \text{ p.s.i.}$$

$$K = 1.5$$

$$\delta = 193,000 \text{ p.s.i.}$$

$$F_{bu} = 296,500 \text{ p.s.i.}$$

$$F_{su} = 119,000 \text{ p.s.i.}$$

(Reference 7,
Table 2.3.1.1(a))

$$M.S. = \frac{1}{\sqrt{R_b^2 + R_s^2}} - 1$$
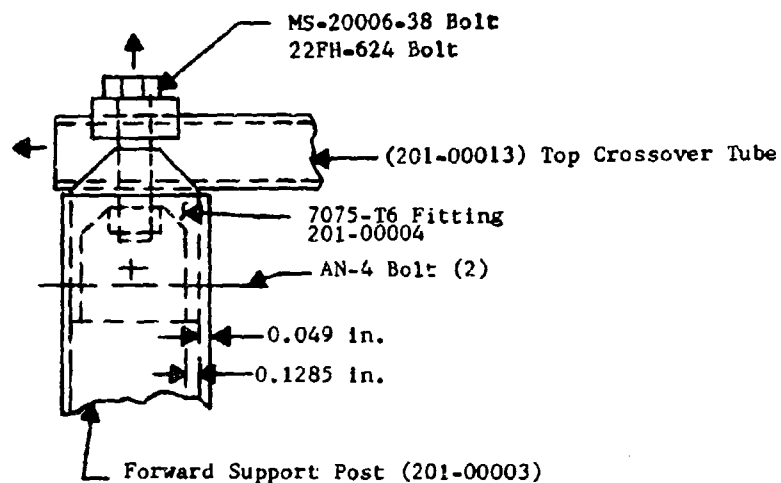
100

where

$$R_b = \frac{f_{bu}}{F_{bu}} = \frac{212,000}{240,000} = 0.883$$

$$R_s = \frac{f_{su}}{F_{su}} = \frac{15,000}{119,000} = 0.126$$

$$M.S. = \frac{1}{\sqrt{0.7959}} - 1 = 0.12$$

<u>Attachment - Forward and Aft Support Posts to Upper Crossover Tube (Reference Drawing 201-000u2)</u>

The critical attachment is for the Forward Support Posts.

AN-4 Bolts

P_t	= -12,201 lb.	Condition 4 Combined Load (see page 59)
P'_{ssu}	= 3,680 lb/bolt	
P'_{bru}	= $F_{bru} A_{br}$	

where

$$A_{br} = Dt_{min} = 0.25 (0.049) = 0.01225 \text{ in.}^2$$

101
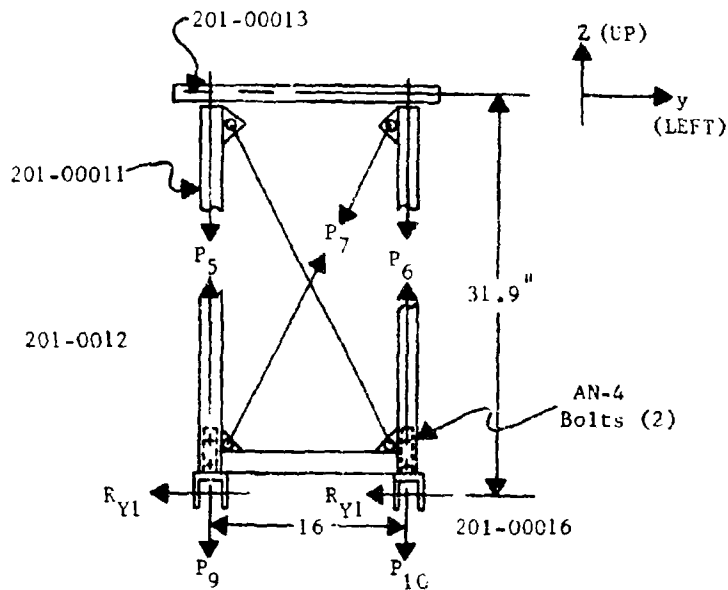
F_{bru} = 251,000 p.s.i. for F_{tu} = 125,000 p.s.i.

(Reference 7)

P_{bru}' = 251,000 (0.01225) = 3,070 lb/bolt

M.S. = $\frac{4\ (3,070)}{12,201}$ -1 = +0.01

AFT SUPPORT POSTS (Reference Drawing 20'-00011)

Figure 31. Aft Support Frame.

The aft support posts are 4:30 steel tubes with:

O.D.	=	1¼ in.
t	=	0.065 in.
A	=	0.242 in.^{2}
D/t	=	19.23
ρ	=	0.4196 in.
F_{tu}	=	150,000 p.s.i.

The maximum compressive load is critical and occurs in Condition 4, Combined Load, and is P_6 = -11,600 lb. (see Table VII).

$$f_c = \frac{P}{A} = \frac{11,600}{0.242} = 47,900 \text{ p.s.i.}$$

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For

$$\frac{L}{P} = \frac{31.9}{0.4196} = 76.2$$

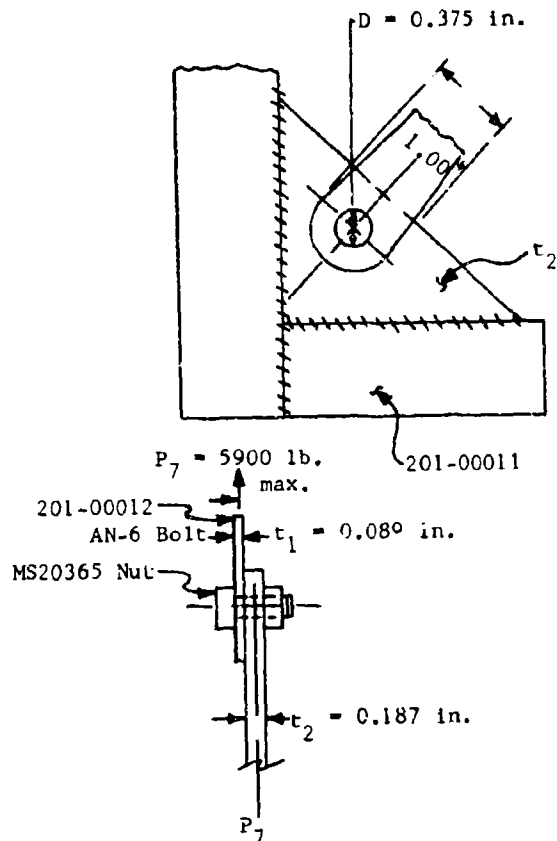$F_c = 51,000$ p.s.i.

(Reference 7,
Figure 2.4.2.3(c))

For

$$\frac{d}{t} = 19.23 < 50, \quad \text{Crippling is not critical,}$$

(Reference 7,
Table 2.4.2.1)

$$M.S. = \frac{F_c}{f_c} - 1 = \frac{51,000}{47,900} - 1 = +0.06$$

LATERAL LOAD SUPPORT ROD (Reference Drawing 201-00012)

Reference Table VII

103

Lug Analysis

$$P'_u = K_{br} A_{br} F_{tu}$$

where

$A_{br} = Dt = 0.375(0.089) = 0.0334$ in.^{2}

$F_{tu} = 150,000$ p.s.i.

To find K_{br},

$\dfrac{a}{D} = \dfrac{0.50}{0.375} = 1.333$

$K_{br} = 1.4$ (Reference 16, Figure 11)

$P'_u = 6,250$ lb.

$P_u = P_7 = 5,900$ lb.

M.S. $= \dfrac{P'_u}{P_u} - 1 = \dfrac{6,250}{5,900} - 1 = +0.06$
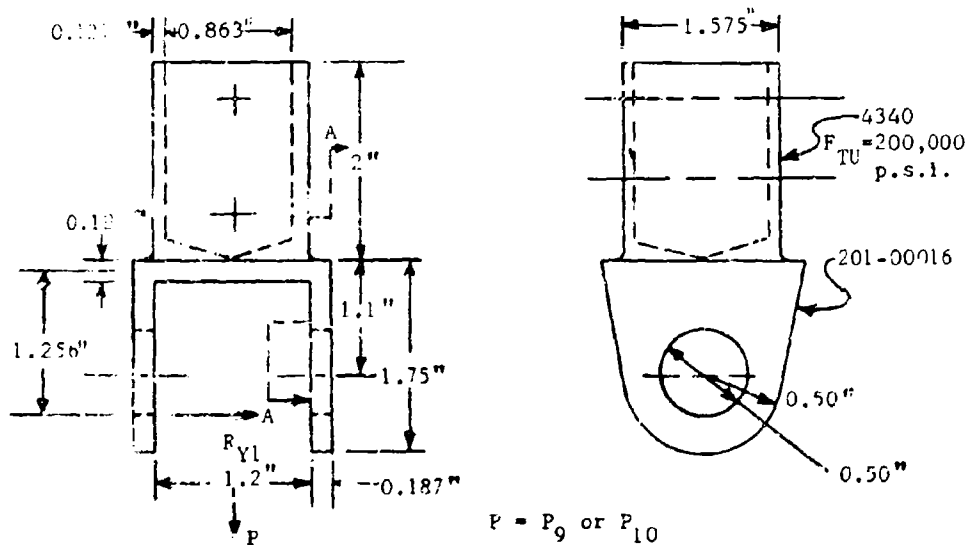
AFT TUBE LOWER FITTING (Reference Drawing 201-00016)

P = P$_9$ or P$_{10}$

Figure 32. Aft Tube Lower Fitting.

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$P = \pm 11,600$ ib. $R_{Y_1} = 1,318$ lb.   Condition 3, Lateral Load

Critical for Section A-A (see Table VII)

Section A-A

$t = 0.187''$ $A = 0.294$ in.^{2}
$\leftarrow 1.575'' \rightarrow$ $I = 0.000854$ in.^{4}
       $\rho = 0.054$ in.

$M = (1.1 - \frac{0.187}{2} + \frac{0.50}{2})$ $1,318 - (- \frac{11,600}{2})(\frac{0.187}{2}) = 1,650 + 542 = 2,192$ in.-lb.

$$f_{bu} = \frac{Mc}{I} = \frac{2,192\ (0.0935)}{0.000854} = 240,000 \text{ p.s.i.}$$

$$f_c = \frac{P}{A} = \frac{11,600}{2(0.294)} = 19,800 \text{ p.s.i.}$$

$$F_{bu} = F_{tu} + (K-1)\ \delta = 200,000 + (1.5-1)\ 193,000$$

(Reference 8,
Table 601:1)

$$= 296,500 \text{ p.s.i.}$$

$$\frac{L}{\rho} = \frac{1.1 - \frac{0.187}{2}}{0.054} = 18.65 < 20; \text{ therefore, use}$$

$$F_{cc} = F_{cy} = 198,000 \text{ p.s.i.} \qquad \text{(Reference 7,}$$
Table 2.3.1.1(a))

Shear stresses are small; therefore, assume that

$$M.S. = \frac{1}{R_b + R_c} - 1$$

where

$$R_b = \frac{f_{bu}}{F_{bu}} = \frac{240,000}{296,500} = 0.809$$

$$R_c = \frac{f_c}{F_{cy}} = \frac{19,800}{198,000} = 0.100$$

$$M.S. = \frac{1}{0.909} - 1 = 0.10$$

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CARRIAGE (Reference Drawing 201-00009)

Condition 1 is the critical design condition (see Table VII for loads).
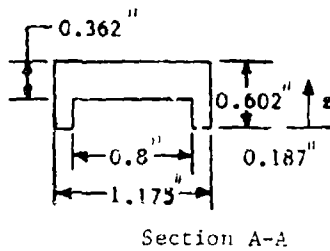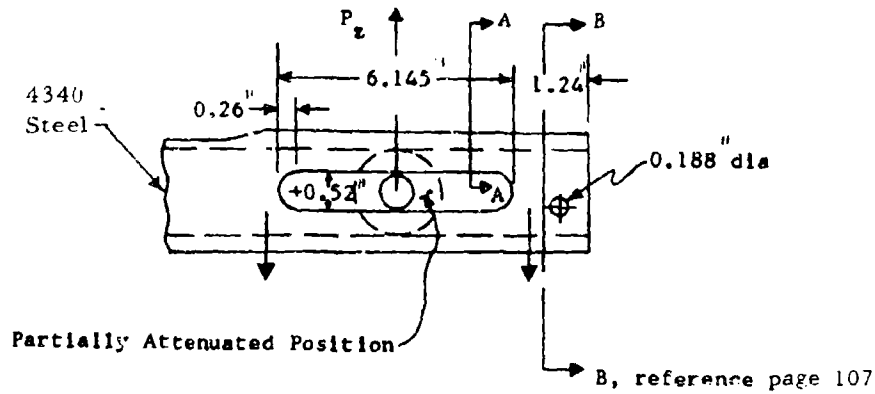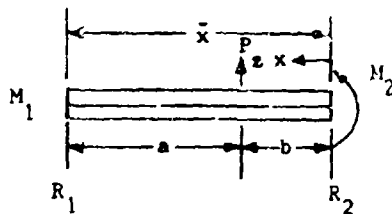

Figure 33. Carriage

$$A = 0.515 \text{ in.}^2$$

$$\bar{z} = 0.379 \text{ in.}$$

$$I = 0.011727 \text{ in.}^4$$

$$\bar{x} = 6.145 - 1/3(0.26)^2 = 5.972 \text{ in.}$$

$$P_z = P_{z1} + \Delta P_z$$

$$P_{z1} = P_9(\sin 85°) = 8,386(\sin 85°)$$

$$= 8,348 \text{ lb.}$$

$\Delta P_z = 535$ lb. (Reference page 110)

$P_z = 8,348 + 535 = 8,883$ lb.

The maximum moment is experienced when

$$x = 1/3 \ L = 1/3 \ (5.972) = 1.657 \text{ in.} = b, \ a = L-b = 4.315 \text{ in.}$$

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M_z = 0.1481(P_z)(5.972)   (Reference 20, Case 32, Table III)

   = 0.1481(8,883)(5.976) = 7,850 in.-lb.

R_2 = $\frac{P_z a^2(3b+a)}{L^3}$ = $\frac{8,883(4.315)^2(4.971 + 4.315)}{(5.972)^3}$ = 7,200 lb.

f_{bu} = $\frac{Mc}{I}$ = $\frac{7,850(0.379)}{0.01173}$ = 253,000 p.s.i.

F_{bu} = F_{tu} + (K-1) δ = 200,000 +(1.5-1) 193,000

   = 296,500 p.s.i.   (Reference 8, Table 601:1)

f_{su} = $\frac{P_z}{A}$ = $\frac{7,200}{0.515}$ = 14,000 p.s.i.

F_{su} = 119,000 p.s.i.   (Reference 7, Table 2.3.1.1(a))

M.S. = $\frac{1}{\sqrt{R_b^2 + R_s^2}}$ -1

where

R_b = $\frac{f_{bu}}{F_{bu}}$ = $\frac{253,000}{296,000}$ = 0.855

R_s = $\frac{f_{su}}{F_{su}}$ = $\frac{14,000}{119,000}$ = 0.118
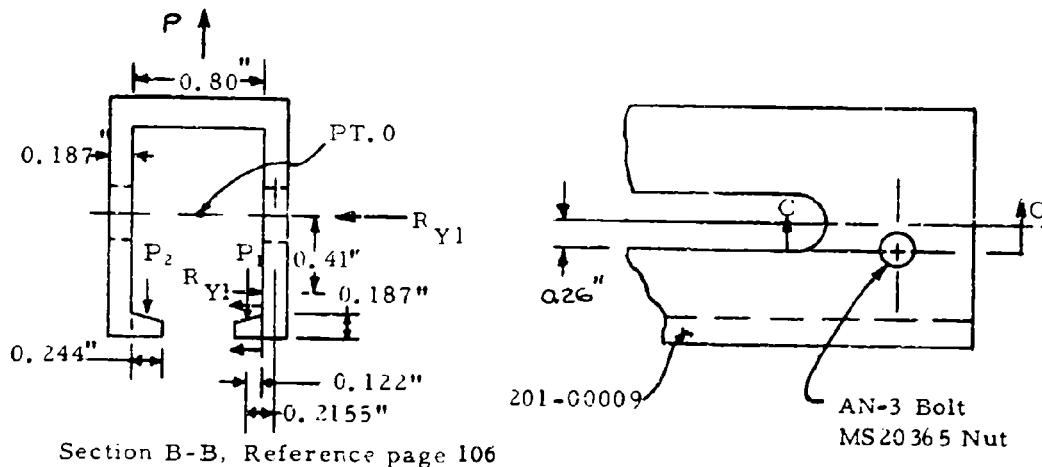
M.S. = $\frac{1}{0.774}$ -1 = + 0.16

Section B-B, Reference page 106

Figure 34.   Carriage - Track Attachment.

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Condition 4, Combined Load, is critical for design

$$P = P_9(\cos 85°) + \Delta P_z = 12,970 \ (\cos 85°) + 0.75(535)$$

$$= 13,301 \text{ lb.}$$

(see Table VII
and page 110)

$$R_{Y1} = 730 \text{ lb.}$$

$$\Sigma F_z = 0, \quad P_1 + P_2 = P = 13,301 \text{ lb.}$$

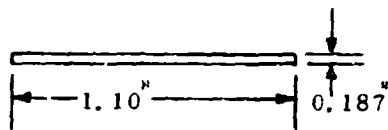$$\Sigma M_o = 0, \quad (P_1 - P_2)(0.80 - 0.244) \ ½ = 730(0.41)$$

$$P_1 = 1,075 + P_2$$

$$2P_2 + 1,075 = 13,301 \text{ lb.}$$

$$P_2 = 6,113 \text{ lb.}$$

$$P_1 = 7,188 \text{ lb.}$$

Section C-C (see page 107)

Effective Section

$$A = 0.206 \text{ in.}^2$$

$$M_{c-c} = 0.864 \left[P_1(0.2155) + R_{Y1}(0.41-0.26) \right]$$

$$= 0.864 \left[7,188(0.2155) + 730(0.15) \right] = 1,435 \text{ in.-lb.}$$

$$f_{bu} = \frac{Mc}{I} = \frac{6M}{bt^2} = \frac{6(1,660)}{1.10(0.187)^2} = 259,000 \text{ p.s.i.}$$

$$f_{tu} = \frac{P_1}{A} = \frac{7,188}{0.206} = 34,900 \text{ p.s.i.}$$

$$f_{su} = \frac{R_{Y1}}{A} = \frac{730}{0.206} = 3,540 \text{ p.s.i.}$$

$$F_{bu} = F_{tu} + (K-1) \ \delta = 296,500 \text{ p.s.i}$$

(Reference 8)

$$F_{su} = 119,000 \text{ p.s.i.}$$

(Reference 7)

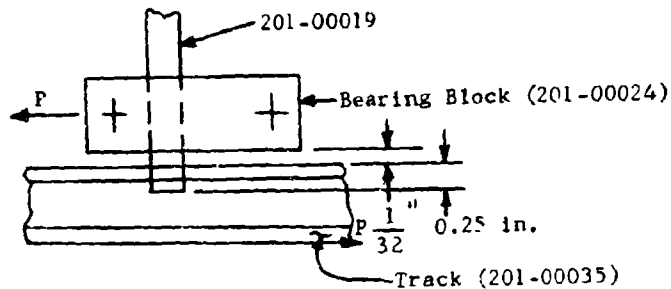$$M.S. = \frac{1}{\sqrt{R_b^2 + R_s^2}} - 1$$

108

where

$$R_b = \frac{f_{bu} + f_{tu}}{F_{bu}} = \frac{293,900}{296,500} = 0.99$$

$$R_s = \frac{f_{su}}{F_{su}} = \frac{3,540}{119,000} = 0.0298$$

$$M.S. = \frac{1}{\sqrt{0.989}} - 1 = +0.01$$

LONGITUDINAL LOCK TO TRACK (Reference Drawing 201-00002)

Movable pins are used to lock the seat to the floor track by engaging holes in each of the two tracks. The pins are the critical items in the installation for Forward Load, Condition 1.

$$P = 8,860/2 = 4,430 \text{ lb. (see page 52)}$$

$$f_{bu} = \frac{Mc}{I} = \frac{4430 \times .14 \times .1565}{0.00047} = 207,000 \text{ p.s.i.}$$

$$f_{su} = \frac{P}{A} = \frac{4,430}{0.077} = 57,500 \text{ p.s.i.}$$

$$F_{bu} = F_{tu} + (K-1)\ \delta = 200,000 + (1.7-1)\ 193,000 = 335,000 \text{ p.s.i.}$$

(Reference 8, Table 601:1)

$$M.S. = \frac{1}{\sqrt{R_b^2 + R_s^2}} - 1$$

where

$$R_b = \frac{f_{bu}}{F_{bu}} = \frac{207,000}{335,000} = 0.62$$

$$R_s = \frac{f_{su}}{F_{su}} = \frac{57,500}{119,000} = 0.483$$

109

$$M.S. = \frac{1}{\sqrt{0.618}} - 1 = +0.28$$

Seat Longitudinal Roller Analysis (Reference Drawing 201-00002)

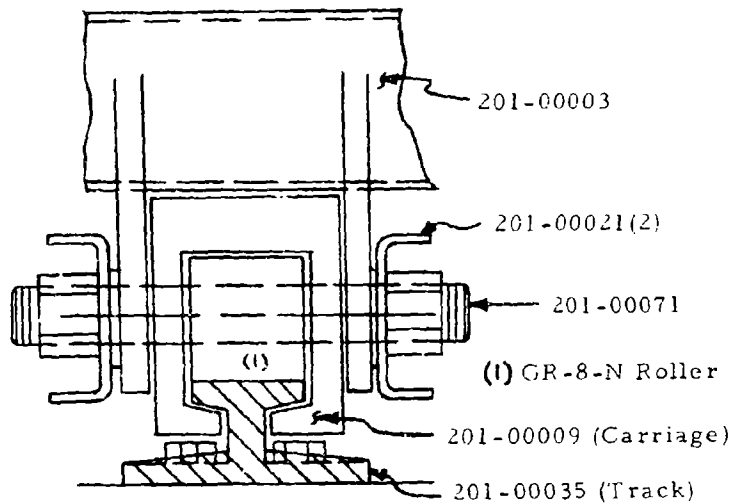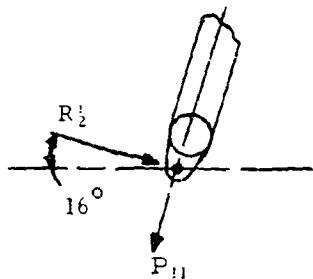Figure 35. Roller - Track Attachment.

For Condition 1, Forward Load:
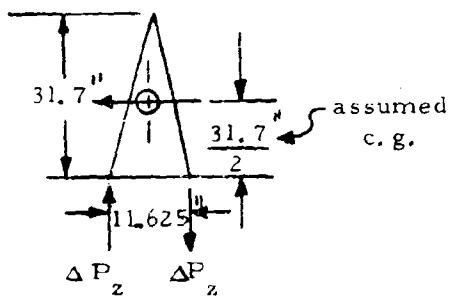

$P_{11} = -8,934$ lb.

(see Table VII)

$R_2' = 877$ lb.

For this condition, the roller load-
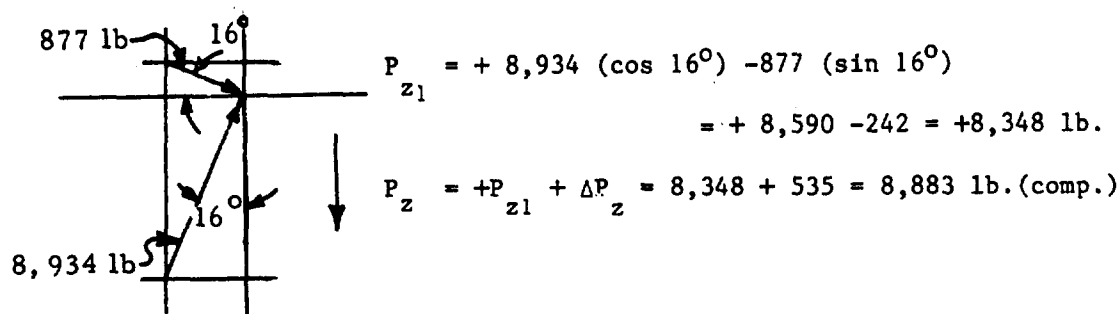ing due to the support structure
weight must also be considered.

Support structure wt. = 35 lb.,
g = 22.5 (see page 51).

$$\Delta P_z = \frac{1}{2}\left[\frac{35(22.5)}{11.625}\left(\frac{31.7}{2}\right)\right] = 535 \text{ lb.}$$

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$$P_{z_1} = + 8,934 \ (\cos 16^\circ) - 877 \ (\sin 16^\circ)$$

$$= + 8,590 - 242 = +8,348 \ \text{lb.}$$

$$P_z = +P_{z_1} + \Delta P_z = 8,348 + 535 = 8,883 \ \text{lb. (comp.)}$$

For Condition 4, Combined Load:

$$P_{11} = 13,259 \ \text{lb.}$$

$$R_2' = \quad 119 \ \text{lb.} \qquad\qquad\qquad \text{(see Table VII)}$$

$$P_z' = +13,259(\cos 16^\circ) - 119(\sin 16^\circ) = 12,750 - 33 = 12,717 \ \text{lb.}$$

$$P_z = P_z' + \Delta P_z = 12,717 + 535(.75) = 13,118 \ \text{lb.}$$

Combined loading incorporates 75 percent of the forward load.

The roller has an allowable normal load of 8,420 pounds, Reference 3, when it is used in conjunction with a track having a hardness of Rockwell "C" 40 and an inner ring-shaft having a hardness of Rockwell "C" 60. This is a load beyond which track brinelling will occur; it is not an ultimate allowable load. The friction coefficient recommended with the proper track arrangement is $\mu = 0.004$. Due to the applied normal load of 8,883 pounds being greater than the allowable normal load of 8,420 pounds, a compensating friction coefficient of $\mu = 0.01$ is assumed. This assumption is based on the knowledge that as brinelling of the track occurs, the friction load that must be overcome to move the roller forward will increase. The effect of the additional load on the roller is considered negligible.

The increased friction coefficient is mentioned only in that the forward attenuator design loads, Condition 1, Table VIII, are dependent upon its value as follows:

$$P_{FWD} = 8,860 \ \text{lb.} \qquad\qquad \text{(see page 52)}$$

Total Normal Load at 4 Support Posts

$$= 4(8,883 \ \text{lb.}) = 35,532 \ \text{lb.} \qquad \text{(see above)}$$

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Total Friction Load Resisting Forward Motion

$$= 0.01(35,532) = 355 \text{ lb.}$$

Therefore, the forward load that must be developed by the forward attenuators is 8,860 -355 = 8,505 pounds.

EVALUATION OF UH-1B HELICOPTER FLOOR UNDERSTRUCTURE FOR INSTALLATION OF HAYES AIRCREW SEAT

The seat attachment loads are applied directly to the main longitudinal beams at BL 14 and BL 30 (reference Figure 36). The BL 30 beams transfer load to the bulkheads at Stations 37, 52, 66 and 78. These bulkheads then in turn transfer loads to the BL 14 beams where they are beamed aft and redistributed throughout the structure supporting the BL 14 beams. It has been determined that Conditions 3 and 4, lateral loading and combined loading, respectively, are critical for floor design (reference Figures 36 and 37). Using the BL 30 beam as an example, load requirements for all floor structure are presented. To present a scope of work that will be required to fully redesign the present floor structure, a summary of suggested modifications is presented herein. The modifications are based on the assumption that all side and forward load components introduced into the floor are reacted by the honeycomb floor aft of Station 66 and that all vertical loads are reacted by the floor structure. Standard structural analysis methods are employed in all calculations. To adequately utilize the present floor design, it is assumed that all compression loads crush the floor structure locally. Only tension loads applied by the seat structure are reacted by the floor structure, except for the BL 14 beams which are required to carry up loads transferred from the BL 30 beams. The lower seat carriage is used to further transmit loads from the seat support posts to the floor structure, as shown in the analysis.
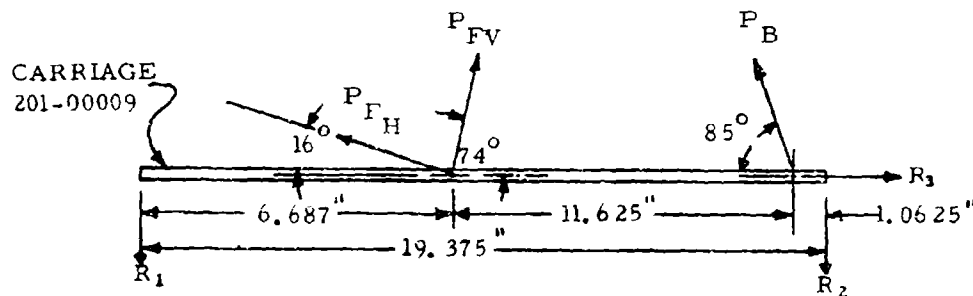
Condition 1 - Forward Loading - Example

where $\qquad P_{11} = P_{12} = -8,934$ lb.

$$P_{F_V} = P_{11} -\cos 16^{\circ}(535) - \tfrac{1}{2}(\sin 16^{\circ})(35)22.5 = -9,546 \text{ lb.}$$

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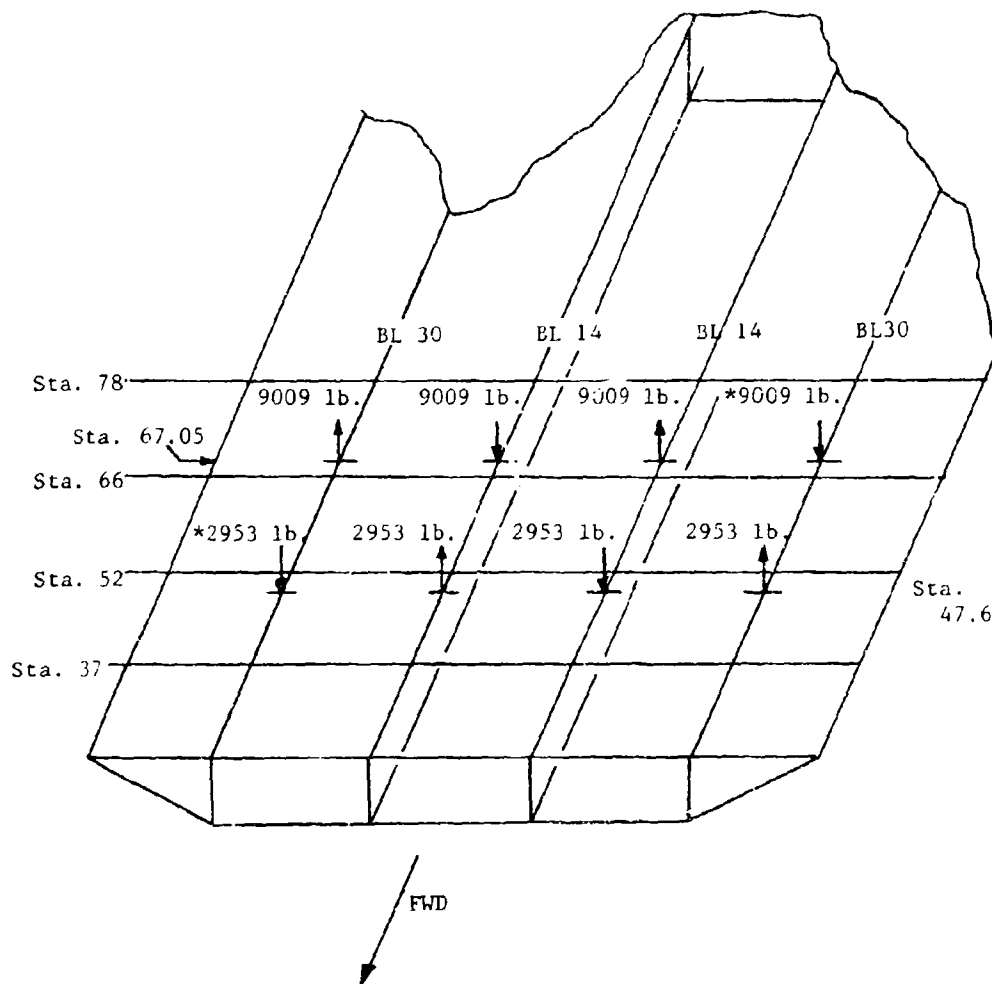
*Denotes local crushing loads

Figure 36. UH-1B Floor Loading, Condition 3.

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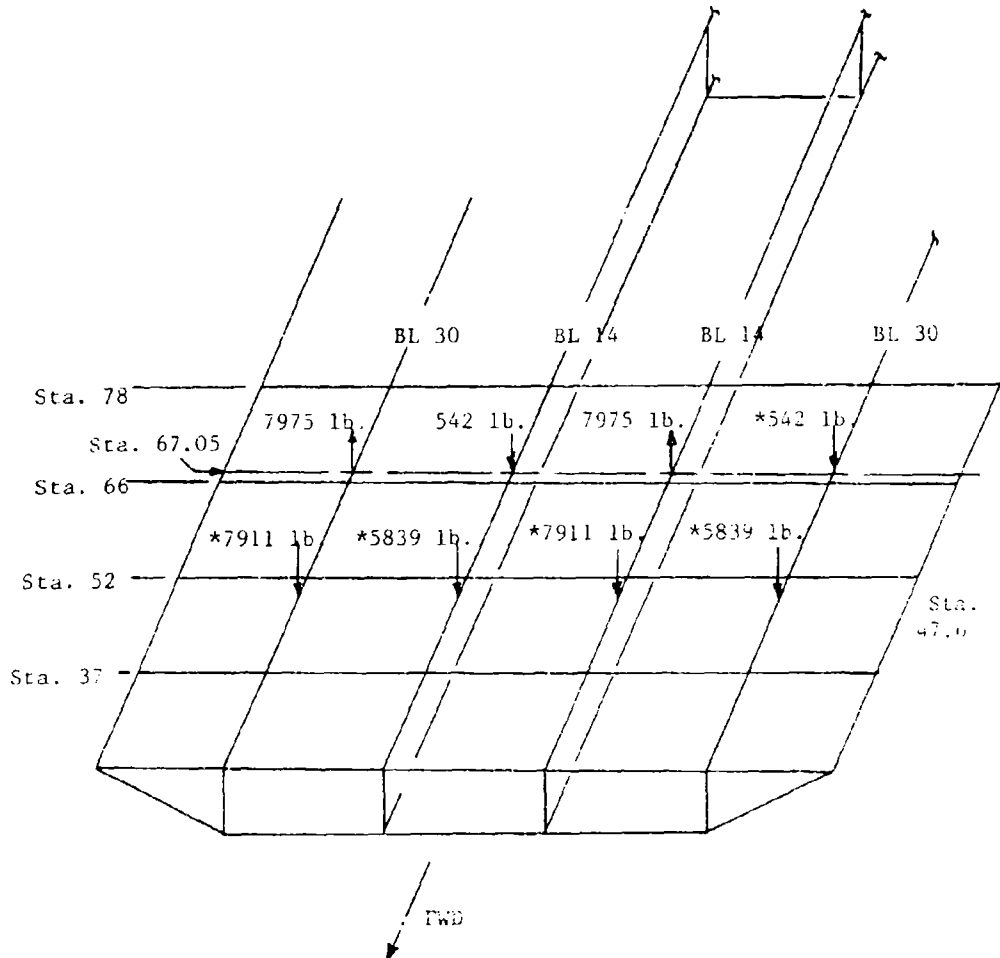
*Denotes local crushing loads

Figure 37. UH-1B Floor Loading, Condition 4.

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where $\quad R'_L = R'_L = 877$ lb.

$$P_{F_H} = R'_L - \sin 16°(535) + \tfrac{1}{2}(35)22.5(\cos 16°) = 1,107 \text{ lb.}$$

where

$$P_9 = P_{10} = 8,836 \text{ lb.}$$

$$P_B = P_9 + 535 = 8,921 \text{ lb.}$$

$$R_1 = \frac{1}{19.375}\left\{\left[\sin 74° (P_{F_V}) + \sin 16° (P_{F_H})\right](11.625 + 1.0625)+ \right.$$

$$\left. \sin 85°(P_B)1.0625\right\}$$

$$R_1 = 0.629(P_{F_V}) + 0.1801(P_{F_h}) + 0.0546 (P_B)$$

$$R_2 = 0.333(P_{F_V}) + 0.096 (P_{F_H}) + 0.941 (P_B)$$

$$R_3 = -0.276(P_{F_V}) + 0.961 (P_{F_H}) + 0.0871 (P_B)$$

$$R_1 = -5,313 \text{ lb.}$$

$$R_2 = +5,313 \text{ lb.}$$

$$R_3 = +4,440 \text{ lb.}$$
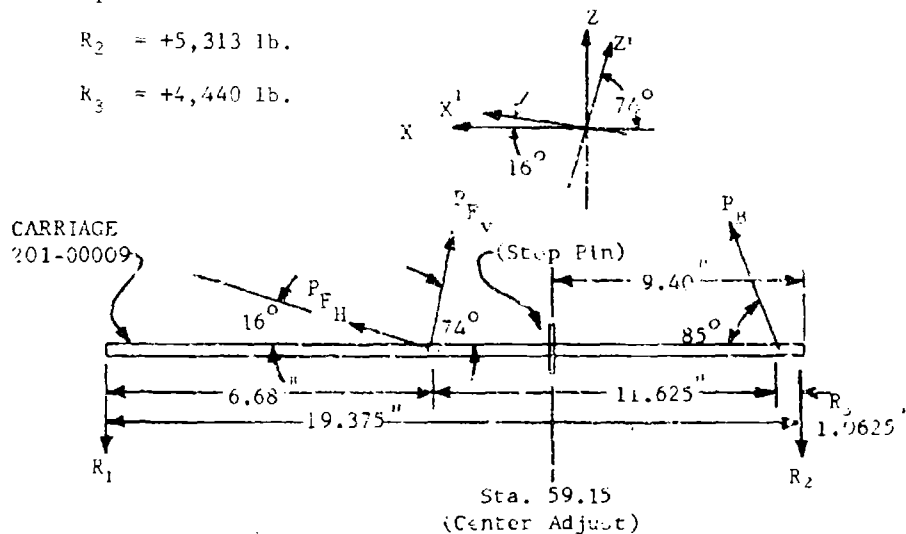
Fore and Aft Seat Adjustment $= \pm 1.5$ inches

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TABLE IX
UH-1B HELICOPTER FLOOR LOADS

	Condition 1 Forward Loading (lb.)	Condition 2 Vertical Loading (lb.)	Condition 3 Lateral Loading (lb.)	Condition 4 Combined Loading (lb.)	Full Fwd Full Side (lb.)
Left Side					
P_{F_V}	- 9,546	- 3,630	+ 6,052	- 9,217	- 5,350
P_{F_H}	1,107	- 1,145	- 1,244	- 1,065	+ 221
P_B	8,921	1,100	-11,600	+ 2,802	- 63
R_1	- 5,313	- 2,436	+ 2,953	- 5,839	- 3,334
R_2	+ 5,313	- 263	- 9,009	- 542	- 1,818
R_3	+ 4,440	0	- 3,873	+ 1,762	+ 1,682
Right Side					
P_{F_V}	- 9,546	- 3,630	- 6,052	-13,726	-13,764
P_{F_H}	1,107	- 1,145	+ 1,244	+ 54	+ 1,993
P_B	8,921	1,100	+11,600	+13,372	+18,893
R_1	- 5,313	- 2,436	- 2,953	- 7,911	- 7,252
R_2	+ 5,313	- 263	+ 9,009	+ 7,975	+13,371
R_3	+ 4,440	0	+ 3,873	+ 5,007	+ 7,355

TABLE X
SEAT ADJUSTMENT POSITIONS

	Fwd Adjust Sta. (in.)	Center Adjust Sta. (in.)	Aft Adjust Sta. (in.)
R_1	47.68	49.18	50.68
R_2	67.05	68.55	70.05

Considering, as a typical calculation, Condition 1, forward loading, with the seat in the forward adjust position, the floor bulkhead loads are determined. All load calculation results are listed in Table XI.

$$P_{37} = - \frac{(52-47.68)}{52-37}(5,313) = -1531 \text{ lb.}$$

$$P_{52} = -3,782 \text{ lb.}$$

$$P_{66} = + \frac{(78-67.05)}{78-66}(5,313) = +4850 \text{ lb.}$$

$$P_{78} = + 463 \text{ lb.}$$
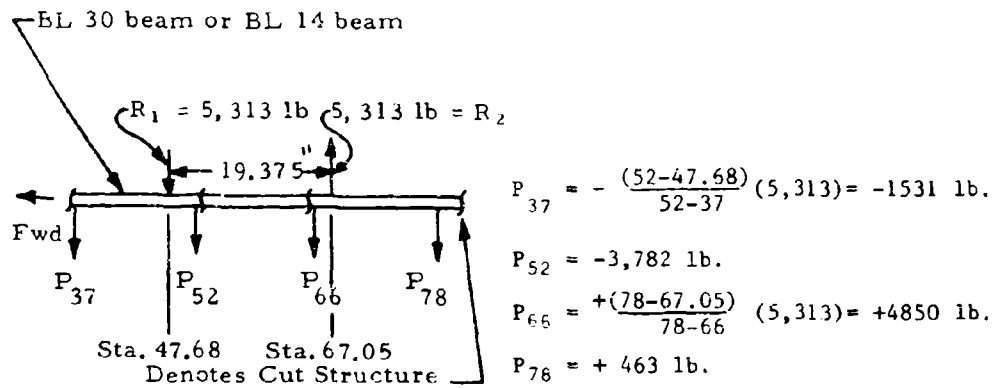
(Reference Drawing 201-00036)

Track Attachment to Floor, BL 30 beam

Maximum aft load = 9,009 lb., Condition 3, Lateral Loading (see Table IX).

$$0.125(\tfrac{2}{3}) + 0.094(\tfrac{1}{2}) = 0.1302 \text{ in.}$$
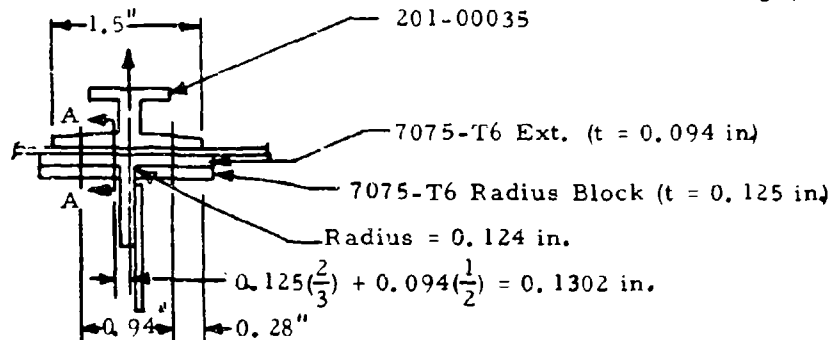
Considering 3 bolts reacting the load at bolt spacing = 1.25 in. (Reference 5, page 21.06).

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$$M_i = \frac{Pl}{8} = 9,009 \frac{(0.94)}{8} = 1059 \text{ in.-lb.}$$

$$t_1 = 0.125 + 0.094 = 0.219 \text{ in.}$$

$$I_{(t = 0.125 \text{ in.})} = \frac{b}{12}(0.125)^3 = 0.00194(\frac{b}{12})$$

$$I_{(t = 0.094 \text{ in.})} = \frac{b}{12}(0.094)^3 = \frac{0.00083(\frac{b}{12})}{0.00277(\frac{b}{12})\text{in.}^4}$$

Moment distribution,

$$M_{(t = 0.125 \text{ in.})} = \frac{0.00194}{0.00277}(1,059) = 741 \text{ in.-lb.}$$

$$M_{(t = 0.094 \text{ in.})} = \frac{0.00083}{0.00277}(1,059) = 318 \text{ in.-lb.}$$

$$M_{A-A} = 1,059 - \frac{9,009}{2}(\frac{0.94}{2} - 0.1302) = -471 \text{ in.-lb.}$$

$$(\text{Tee}) f_{bu} = \frac{Mc}{I} = \frac{6M}{bt^2} = \frac{6(471)}{3.75(0.094)^2} = 85,500 \text{ p.s.i.}$$
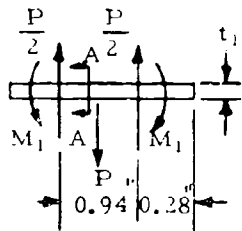
$$(\text{Radius Block}) f_{bu} = \frac{Mc}{I} = \frac{6M}{bt^2} = \frac{6(741)}{3.75(0.125)^2} = 76,200 \text{ p.s.i.}$$

$$F_{bu} = F_{tu} + (k-1)\, \delta = 78,000 + (1.5 - 1)\, 73,000$$

$$= 114,000 \text{ p.s.i., } f_{bu} = 85,000 \text{ p.s.i.}$$

(Reference 7,
paragraph 601.22)

$F_{bu} > f_{bu}$ and shear stresses are small; therefore, present structure is acceptable.

Bolt load, $P_{tu} = \frac{P}{2} +$ Load due to the Tee Flange Deflection

$$= \frac{1}{3}\left[\frac{9,009}{2} + \frac{M_i}{0.28}\right] = 2,758 \text{ lb.}$$

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TABLE XI

UH-1B HELICOPTER FLOOR BULKHEAD LOADS

Bulkhead Sta.	Condition 1 Forward Loading		Condition 2 Vertical Loading		Condition 3 Lateral Loading		Condition 4 Combined Loading		Full Fwd Full Side	
	Fwd Adj. (lb.)	Aft Adj. (lb.)	Fwd Adj. (lb.)	Aft Adj. (lb.)	Fwd Adj. (lb.)	Aft Adj. (lb.)	Fwd Adj. (lb.)	Aft Adj. (lb.)	Fwd Adj. (lb.)	Aft Adj. (lb.)
Left Side										
37	-1,531	- 467	(1)		+	+ 222	-1,680	- 513	- 962	- 293
52	-3,782	-4,840			+2,225	+2,850	-4,159	-5,370	- 2,372	- 304
66	4,850	+3,520			-8,220	-5,960	505	- 366	- 1,660	-1,200
78	463	+1,900			- 789	-3,240	37	- 152	- 158	- 650
Right Side										
37	-1,531	- 467	(1)		- 728	- 222	-2,275	- 695	- 2,090	- 638
52	-3,782	-4,820			-2,225	-2,850	-5,736	-7,350	- 5,162	-6,610
66	4,850	+3,520			+8,220	+5,960	+7,280	+5,280	+12,200	+8,850
78	463	+1,900			+ 789	+3,240	+ 695	+2,860	- 1,171	+4,820

(1) Not critical by comparison with other load conditions

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Bolt - NAS 1588, $\frac{3}{16}$ dia, P'_{tu} = 3,600 lb.

Nut   KAYLOCK K1400, P'_{tu} = 3,470 lb.

Therefore, the bolt-nut combination is satisfactory.

By Reference 5, this track attachment location is critical for all aircraft after manufacturer's serial number 274.

BL 30 Beam

The maximum moment occurs at Sta. 70.05 with the seat in the aft adjust position, lateral loading, Condition 3.

All material is 7075-T6 aluminum

R_2 = 9,009 lb
4.05"
8.78"

4.75-dia. flanged lightening hole
Drawing 204-031-710 of Reference 5

P_{66} = 5,960 lb P_{78} = 3,240 lb

M_M = 5,960(4.05) = 24,200 in.-lb

2.1"
BL 30
t = 0.025 in.
W. L. 22.00
0.72
0.47
t = 0.094 in.
t = 0.050 in.
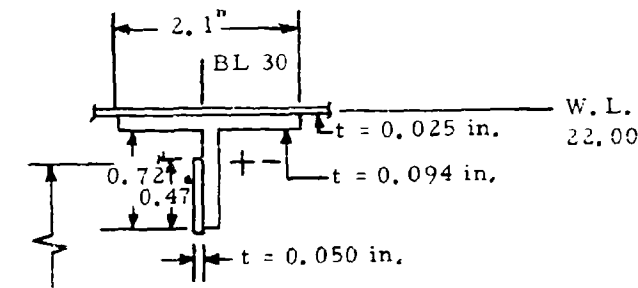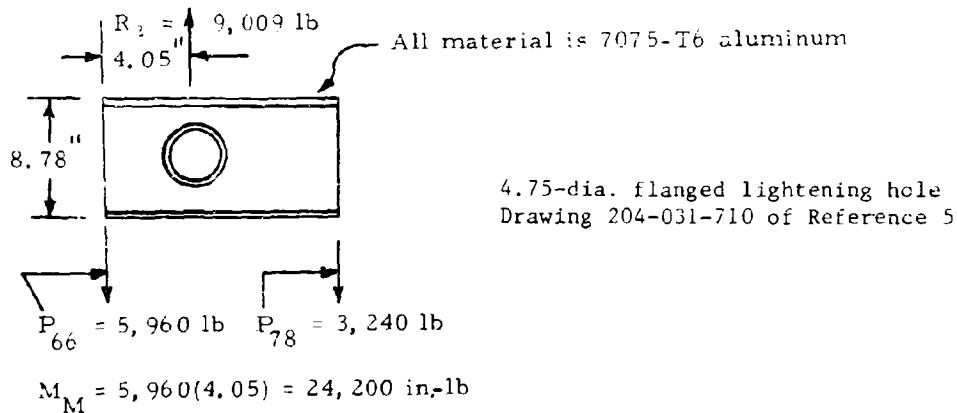
h = 8.0"
assumed

Upper chord area = 2.1(0.094 + 0.025) + 0.72(0.094)+0.47(0.050)

$$= 0.3412 in.^2$$

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$h = 8.0''$
assumed
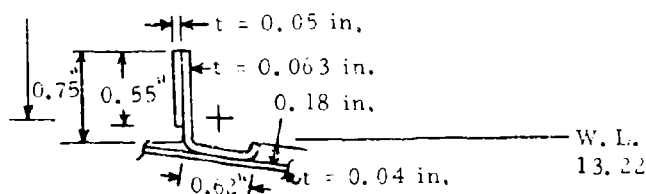
Lower chord area $= (0.75 + 0.62 + 0.18)0.063 + 0.55(.05) + 1(0.04)$

$$= 0.165 \text{ in.}^2$$

Load in lower chord, $P = -\dfrac{M}{h} = -\dfrac{24,200}{8} = -3,025 \text{ lb.}$

Average stress in the lower chord, $f_c = \dfrac{P}{A} = -\dfrac{3,025}{0.165} = -18,350 \text{ p.s.i.}$ compression.

$$\frac{b}{t} = \tfrac{1}{2}(\frac{1.0}{0.04}) = 12.5$$

Cladding thickness is 4 percent per side.

$$\frac{b}{t} = \frac{12.5}{1-(0.04)^2} = 13.6, \text{ one edge fixed, 7075-T6 material}$$

$$F_{cc} = 38,000 \text{ p.s.i.}$$

(Reference 8,
Figures 403:18
and 403:19)

The upper and lower chord margins of safety are large. The web, stiffener, and attachment designs are impractical to perform at this time due to the present lightening hole utilization. Complete helicopter drawings are required to determine whether a lightening hole design would be the lightest.

Bulkheads 37, 52, 66 and 78

The allowable bulkhead loads taken from Reference 5, page 21.04 are

$$P_{37} = \pm 1,306 \text{ lb.}$$

$$P_{52} = \pm 946 \text{ lb.}$$

$$P_{66} = \pm 3,472 \text{ lb.}$$

$$P_{78} = \pm 3,900 \text{ lb.}$$

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The load requirements of the bulkheads are

$$P_{37} = +\ 728\ \text{lb.}$$

$$P_{52} = +\ 2850\ \text{lb.} \qquad \text{(see Table XI)}$$

$$P_{66} = +\ 8220\ \text{lb.}$$

$$P_{78} = +\ 3240\ \text{lb.}$$

Bulkheads 52 and 66 must be uprated from 946 pounds to 2850 pounds and 3472 pounds to 8220 pounds, respectively, to accommodate all loading conditions except the full forward and full side condition. To consider the full forward and full side condition, all bulkheads except 37 would have to be uprated to

$$P_{52} = +\ 6610\ \text{lb.}$$

$$P_{66} = +\ 12,200\ \text{lb.} \qquad \text{(see Table XI)}$$

$$P_{78} = +\ 4820\ \text{lb.}$$

The full forward and full side condition would necessitate a complete floor redesign with all major structure replaced. Preliminary analysis has shown that the remaining load conditions can be sustained by adding structure or by replacing structure that can be replaced with considerably less trouble than a complete redesign. Therefore, all major floor structure analysis is performed only for the four primary load conditions.

The following structural changes and additions are suggested for bulkhead 52 as a result of similar analyses as presented for the BL 30 beams (Reference Drawing 201-00036):

- Add an 0.08-x-0.7-x-0.7-inch 7075-T6 angle to the upper chord.

- Utilize 0.05-x-0.75-x-0.75-inch 7075-T6 angle stiffeners to replace present stiffeners.

- Utilize an 0.04-inch 7075-T6 web to replace the present web.

- Add an additional heavy angle fitting (t = 0.25 in. 7075-T6) for upper chord splicing through the BL 14 beams; for attachments, use 3-3/16-inch-diameter steel Hi-Shear rivets.

- Utilize for web attachments:

  - Web to chords - DD6 rivets at present spacing.

  - Web to stiffeners - AD5 rivets at 1-1/8-inch spacing.

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Employing the same procedure for bulkhead 66, the following changes are suggested:

- Utilize the present 0.072-inch tee section as the upper chord (Alcoa Die No. 26846), add an 0.09-x-0.75-x-1.25-inch 7075-T6 angle to the upper chord.

- Utilize 0.08-x-0.75-x-0.75-inch 7075-T6 angle stiffeners to replace present stiffeners.

- Utilize an 0.072-inch 7075-T6 web to replace the present web.

- Add an additional heavy angle fitting (t = 0.25 in. 7075-T6), thus providing one on either side of the bulkhead 66 web for upper chord splicing through the BL 14 beams; for attachments, use 3-¼-inch-diameter steel Hi-Shear rivets in double shear.

- Utilize for web attachments:

  - Web to chords - 3/16-inch steel Hi-Shear rivets at present spacing.

  - Web to stiffeners - DD6 rivets at 1-1/8-inch spacing.

A procedure similar to that used for the bulkheads is applied to the BL 14 beam loads, reference Figures 35 and 36, resulting in the following recommendations:

Station 37 to Station 66

- Utilize 0.04-x-0.625-x-0.625-inch 7075-T6 stiffeners to replace present stiffeners.

- Utilize an 0.025-inch 7075-T6 web to replace the present web.

- Utilize AD5 rivets for web to chord and web to stiffener attachments at present spacing.

Station 66 - Aft

- Utilize 0.063-x-0.875-x-0.875-inch 7075-T6 stiffeners to replace present stiffeners.

- Utilize an 0.063-inch 7075-T6 web to replace the present web.

- Utilize 3/16-inch steel Hi-Shear rivets for web to chord and web to stiffener attachments at

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present spacing; aft of Station 71, use DD6 rivets
for these attachments.

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APPENDIX II. ENERGY ABSORPTION

Kinetic energy which must be absorbed during deceleration can be absorbed by a number of different devices. The concept used on the Hayes-designed crash seat involves repeated, reversed bending of a wire or rod by pulling it around a spindle or series of spindles. As a result of preliminary testing and analysis, the energy absorption device studied appears to be ideally suited for the crash seat application and other applications, particularly where low inertia load and long deceleration stroke are desirable. It appears that the wire bending concept should be limited only by a pull force equal to wire yield stress times cross-sectional area. The theoretical upper limit potential for unit energy absorption for a wire with yield stress of 125,000 p.s.i. and density of .283 per pound per cubic inch is: Energy per pound = $125,000/.283(12) = 36,000$ foot-pounds per pound. This potential is greater than any of the proven concepts known to the writer and could conceivably be exceeded with higher yield or lower density material. Desirable wire material properties in addition to high yield strength and light weight are low notch sensitivity, ductility, and capacity for alternating (tensile and compressive) plastic straining or cold working.

The basic principle of internal energy absorption through reversed bending lies in the hysteresis developed by the stress-strain curve as illustrated below.
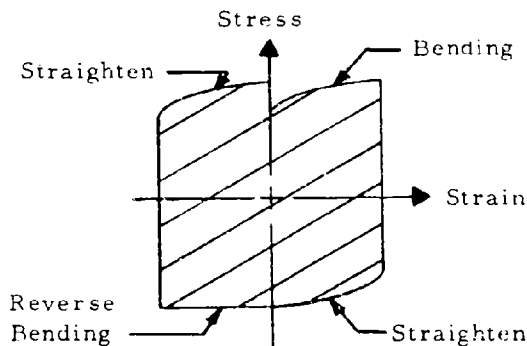
The area bounded by the stress-strain curve multiplied by the respective affected volume of material gives a measure of internal energy absorption.

The total energy absorption includes internal energy absorption plus friction energy. Friction energy is the summation of friction drag force between moving parts multiplied by relative travel distance of adjacent parts.

INTERNAL ENERGY ABSORPTION

Given a solid circular rod of diameter, d, it is desired to find internal energy absorption (U_I) when the rod is bent from a straight axis to a mean

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radius of curvature, r_c:
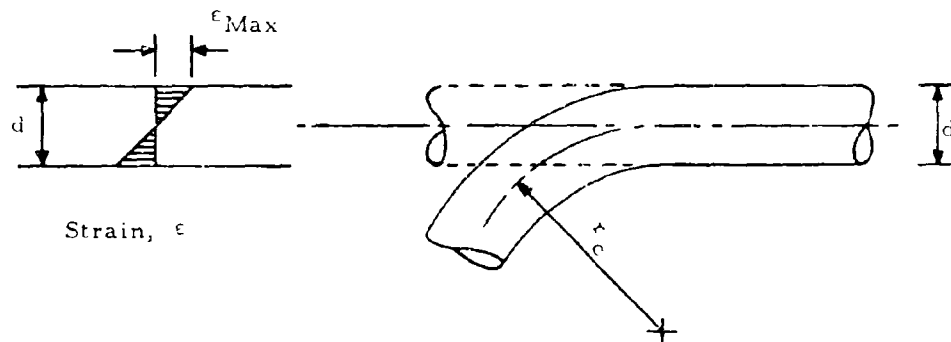
Strain, ϵ

Maximum fiber strain (ϵ_{max}) is given by the following relation:

$$2\epsilon_{max} = \pm \frac{\pi (r_c + \frac{d}{2}) - (r_c - \frac{d}{2})}{\pi r_c}$$

solving

$$\epsilon_{max} = \pm \frac{d}{2r_c}$$

where r is rod radius, $d = 2r$, and

$$\epsilon_{max} = \pm \frac{r}{r_c}$$

ROD CROSS SECTION

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Strain at Y, $\epsilon = \dfrac{Y}{r} \epsilon_{max}$

$$\epsilon = \dfrac{Y}{r_c}$$

Differential area at Y, $dA = 2 \, X \, dY$

$$X = \sqrt{r^2 - Y^2}$$

$$dA = 2\sqrt{r^2 - Y^2} \; dY$$

Differential energy per unit length of rod, where σ = stress,
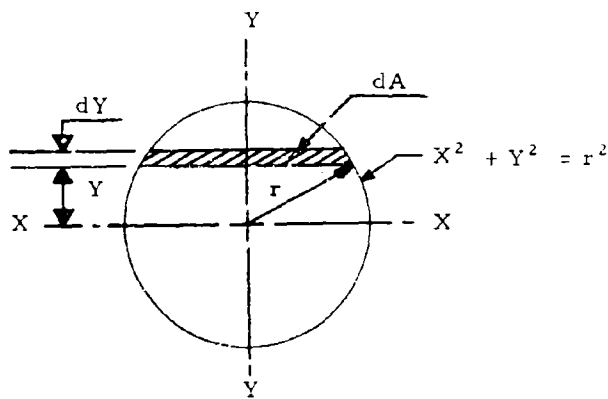
$$dU_I = \sigma \, \epsilon dA; \text{ i.e., Stress x Strain x Volume}$$

$$\dfrac{dU_I}{dY} = 2 \, \sigma \, \epsilon \, \sqrt{r^2 - Y^2}$$

By integration, we find the solution for each wire curvature from straight axis to mean radius, r_c, or from mean radius, r_c, to straight axis. For unit length of rod,

$$U_I = \int_{-r}^{r} \sigma \, \epsilon dA$$

or, substituting, $dA = 2\sqrt{r^2 - Y^2} \; dY$

$$U_I = 4 \int_{o}^{r} \sigma \, \epsilon \, \sqrt{r^2 - Y^2} \; dY$$

Observing that stress, U_I, is a function of Y and is derived from the material stress-strain curve in the plastic range, mathematical representative of σ is inconvenient. We therefore utilize a graphic integration by plotting $\dfrac{dU_I}{dY}$ versus Y and determining enclosed area as follows:

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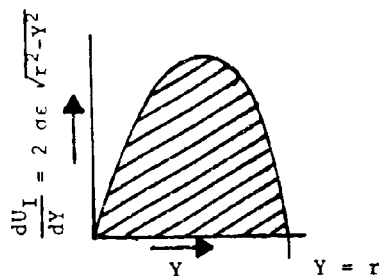
Internal energy for each wire curvature, U_I, is two times the shaded area.

ALTERNATE SOLUTION FOR INTERNAL ENERGY WITH IDEALIZED STRESS-STRAIN CURVE

In the case where stress-strain curve can be idealized as a straight line such that stress, $\sigma = \sigma_o + E_p$,

σ_o = stress at strain equal zero

E_p = slope of plastic S-S curve

ϵ = strain

the following solution can be applied with greater facility than the graphic solution.

INTERNAL ENERGY SOLUTION WITH IDEALIZED STRESS-STRAIN CURVE

$$U_I = 4 \int_0^r \sigma \epsilon \sqrt{r^2 - Y^2} \; dY \qquad (4)$$

$$\epsilon = \frac{Y}{r_c}$$

where plastic stress-strain curve can be represented as straight line, e.g., high yield material.

$$\sigma = \sigma_o + E_p \epsilon$$
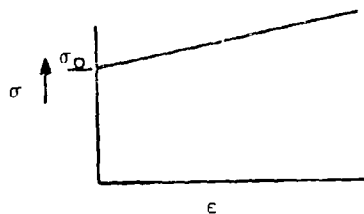
Equation (4) becomes

$$U_I = 4 \int_0^r (\sigma_o + E_p \epsilon) \; \epsilon \; \sqrt{r^2 - Y^2} \; dY$$

$$= 4 \int_0^r (\sigma_o \epsilon + E_p \epsilon^2) \; \sqrt{r^2 - Y^2} \; dY$$

$$= \frac{4}{r_c} \int_0^r (\sigma_o Y + \frac{E_p}{r_c} Y^2) \sqrt{r^2 - Y^2} \; dY$$

$$U_I = \frac{4}{r_c} \int_0^r (\sigma_o Y \sqrt{r^2 - Y^2} + \frac{E_p}{r_c} Y^2 \sqrt{r^2 - Y^2}) \; dY$$

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Integrating, we get the solution for use with the idealized stress-strain curve,

$$U_I = \frac{4\sigma_c \, r^3}{3r_c} + \frac{\pi \, r^4 Ep}{4r_c}$$

FRICTION ENERGY

Friction energy, U_F, is the product of frictional drag times distance. Drag froces are dependent on spindle and guide orientation as well as on friction coefficient. Because of the many variable arrangements which may be used in functional absorber concepts, no rigorous general solution is attempted at this stage of analysis. Friction energy is represented generally for unit wire travel distance as

$$U_F = \Sigma\mu F$$

where

$$\mu = \text{friction coefficient}$$

$$F = \text{force normal to wire axis}$$

Following is a typical detailed solution for friction energy analysis:

$$M_p = \text{plastic moment}$$

$$M_p = U_I/\text{bend or } \tfrac{1}{2} \, U_I/\text{spindle}$$

ΣMOMENTS

$$U_T = 2M_p + \mu R_V \frac{D}{2} + \mu R_H \frac{D}{2} + \mu \frac{M_p \frac{D}{2}}{a_i} \text{ (Approx.)} \tag{5}$$
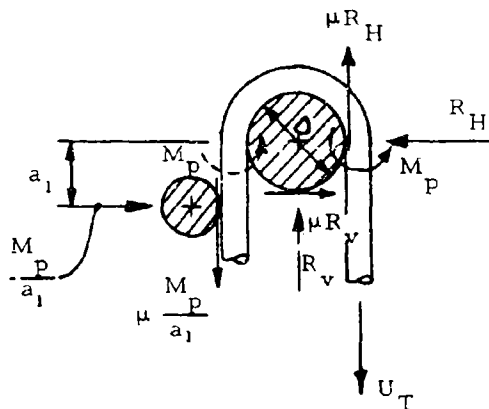
ΣHORIZONTALS

$$R_H = \mu R_V + \frac{M_p}{a_1}$$

$$R_V = \frac{R_H - \frac{U_I}{2A}}{\mu} \tag{6}$$

ΣVERTICALS

$$U_T = R_V + \mu R_H - \mu \frac{M_p}{a_1} \tag{7}$$

Noting $U_I = 2M_p$ and substituting,

$$U_T = U_I + \mu \frac{D}{2} \left[R_V + R_H + \frac{U_I}{2a_1} \right] \tag{8}$$

Substituting (6) into (8),

$$U_T = U_I + \frac{D}{2} \left[(1 + \mu)R_H - (1 - \mu)\frac{U_I}{2a_1} \right]$$

$$U_T = \left[1 - \frac{D(1-\mu)}{4a_1} \right] U_I + \frac{D(1+\mu)}{2} R_H \tag{9}$$

Substituting (6) into (7),

$$U_T = \frac{1}{\mu} R_H + \mu R_H - \mu\frac{U_I}{2a_1} - \frac{1}{\mu} \frac{U_I}{2a_1}$$

$$U_T = \frac{\mu^2 + 1}{\mu} R_H - \frac{\mu^2 + 1}{\mu} \frac{U_I}{2a_1} \tag{10}$$

$$R_H = \frac{\mu}{\mu^2 + 1} \left[U_T \right] + \frac{U_I}{2a_1}$$

Substituting (10) into (9),

$$U_T = \left[1 - \frac{D(1-\mu)}{4a_1} \right] U_I + \left[\frac{D(1+\mu)}{2} \frac{\mu}{\mu^2 + 1} \right] U_T$$

$$+ \left[\frac{D(1 + \mu)}{4a_1} \right] U_I$$

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$$U_T = \left[\frac{1 + \frac{\mu D}{2a_j}}{1 - \frac{D}{2} \frac{u + \mu^2}{1 + \mu^2}} \right] U_I$$

Numerical check of representative examples reveals that U_T can be approximated within $\pm$ 1-percent by the following relation where $\mu \leq .20$ and $D = .625$ inches:

$$U_T \simeq (1 + \mu) U_I$$

However, additional work is necessary to justify general use of the above approximation.

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PRELIMINARY TESTING AND EVALUATIONS

Preliminary testing has been performed in two phases. The initial test
phase was to verify concept feasibility and to select materials for further
study. Both single and double spindle concepts were tested in preliminary
tests utilizing three rod sizes: 1/8-inch-diameter 4130 steel, 3/32-inch-
diameter beryllium-copper alloy, and 3/16-inch-diameter AISI 316 stainless
steel. In the single spindle tests, the rod travelled around one spindle
approximately 360°, experiencing one bending and one straightening deforma-
tion. In the double spindle test, the rod travelled around two spindles so
as to experience one bending, one reverse bending, and two straightening
deformations. Calculated internal energy, test results and additional rod
characteristics are listed on Table XII.

On the basis of the test results and study, 4130 steel heat treated to
150,000 p.s.i. ultimate was selected for preliminary sizing and development
of functional components for the Hayes seat design.

The preliminary development test phase is designed to accumulate data for
analysis refinement and more accurate sizing of functional attenuator com-
ponents using 4130 steel rods. Rod sizes of 5/32-, 3/16- and 9/32-inch
diameter were selected for testing. The test variables include wire dia-
meter, d, and ratio of wire to spindle diameters, d/D. The program should
also provide data on the facility of achieving a specific heat treat value
(150,000 p.s.i. ultimate) and data on fatigue life in the very low cycle
range. (Even through the attenuators are one-operation devices, it is
expedient to ascertain that suitable safety margins are provided in order
to assure functional reliability.)

Preliminary development test to date has included slow pull tests and
dynamic testing of load attenuators to be installed in the UH-1B seat de-
signed under Bell Helicopter Company Contract DA 44-177-AMC-89(T) with
USATRECOM*, Fort Eustis, Vriginia. The UH-1B seat was prepared for drop
testing at Aviation Safety Engineering and Research, Phoenix, Arizona, for
the primary purpose of studying the Hayes load attenuator under crash sim-
ulating load conditions. Details of the drop tests are given in Reference
13, AvSER Report No. M-66-16, and are further discussed in the following
section. The load attenuators tested consisted of 5/32-inch-diameter rods
of 4130 steel heat treated to 150,000 p.s.i. ultimate. Each rod was pulled
180° around a 5/8-inch spindle. Results of the slow pull and dynamic tests
are given in Figures 38 and 39. Pull tests are evaluated in the following
paragraph, and load rating is established. It is apparent that future de-
signs should afford more consistent friction resistance in order to permit
load prediction within a less narrow margin than the ±6 percent exhibited
by the specimens tested.

During development of the attenuators for installation in the UH-1B seat
and drop testing, it was demonstrated that pull rod physical characteristics

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*Now U. S. Army Aviation Materiel Laboratories.

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TABLE XII
LOAD ATTENUATOR DATA SUMMARY,
INITIAL TEST, AND ANALYSIS

Test Number	1A	1B	2A	2B	3A	3B
Rod Diameter (in.)	.124		.156		.187	
Material	4130 Steel		Beryllium-Copper Alloy		316 Stainless Steel	
Tensile Yield Stress/1000	130	130	105	105	90	90
Ultimate Stress/1000	146	146	115	115	108	108
Spindle Diameter (in.)	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
Rod Strain, ε_{max}	.198	.198	.238	.238	.274	.274
$\Sigma\varepsilon_{max} = \varepsilon_{max} \times$ number bends	.396	.792	.476	.952	.548	1.096
Internal Energy, U_I (lb.)	289	578	432	864	660	1,320
Test Pull Force (Ave.) (lb.) U_T	350	710	470	970	800	1,540
Friction Coefficient (Ave.) $(U_T - U_I) / U_I$	.21	.23	.09	.12	.21	.17
Test Unit Energy Absorption (ft.-lb/lb) U_T/wgt. per ft.	14,200		12,200		14,200	
Total Energy Absorption Potential (ft.-lb/lb) Yield Stress/Density	38,300		28,200		26,200	

can easily be duplicated in separate heat and draw operations. Four different heat treatment lots exhibited ultimate tensile strengths between 149,000 p.s.i. and 150,000 p.s.i.

Safety margin in total strain was demonstrated for the UH-iB seat attenuator replacement by pulling a pair of rods over the spindle for six successive times with no visible evidence of wire damage.
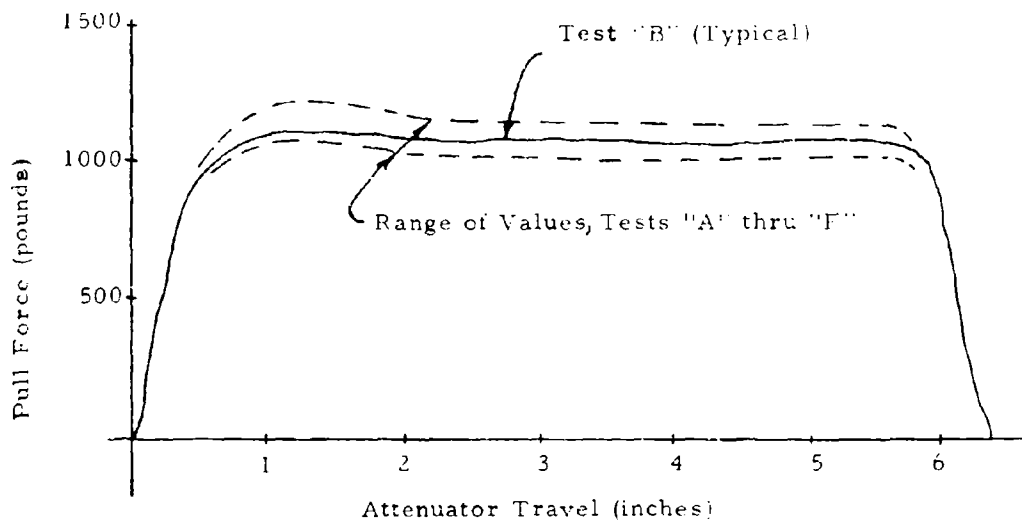

Figure 38. Load Attenuator Slow Pull Test Results
(2 Rods, 5/32-Inch-Diameter Single Spindle,
.625-Inch-Diameter).
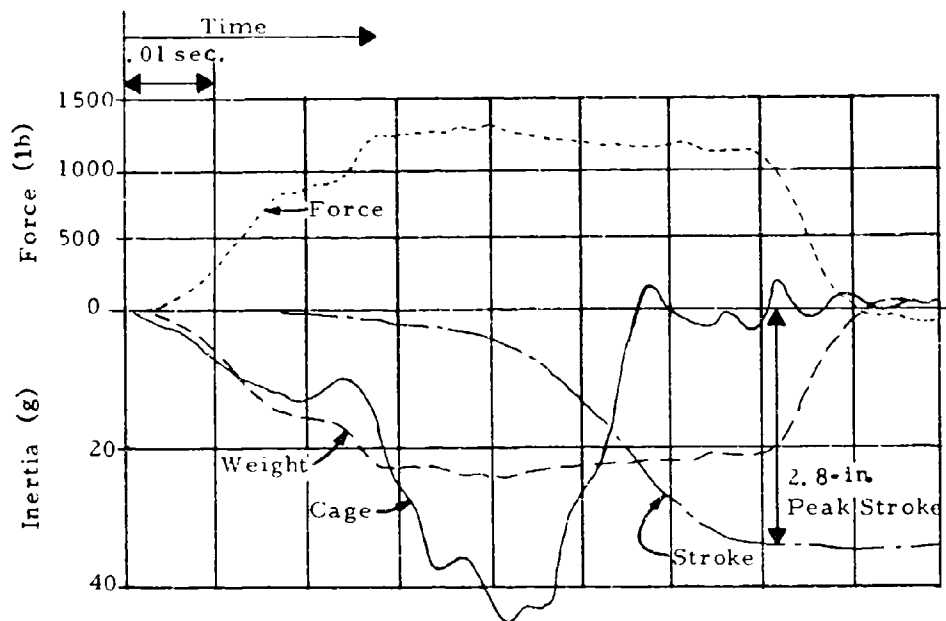
Figure 39. Load Attenuator Calibration Drop Test
(2 Rod, 5/32-Inch-Diameter, 4130 Steel,
150 k.s.i. ult. Single Spindle, .625-Inch
Diameter, 57-lb. Wt.).

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TABLE XIII
RESULTS OF LOAD ATTENUATOR PULL TESTS

Rod: Diameter = .1552 in.
4130 Steel, Yield = 141.5, Ult = 150

Spindle: Diameter = .625 in.

Rate of Pull: Approximately 1 ft. per min.

TEST RESULTS
(2 Rods Per Unit)

TEST	PULL FORCE START (lb.)	RUN (lb.)	
A	1,160	1,110	
B	1,140	1,100	Spindle rotated in fixture
C	1,120	1,050	
D	1,100	1,030	
E	1,240	1,170	Spindle did not turn, rod
F	1,220	1,150	slid on spindle
AVE.	1,163	1,098	

The data variation of the pull force was ±6 percent for starting and ±6.4 percent for running.

Determination of friction coefficient,

$$U_I = 954 \text{ lb. (calculated)}$$

$$U_T = (1 + \mu) \, U_I$$

$$\mu = \frac{U_T - U_I}{U_I}$$

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Start μ = .217

Run μ = .151

Variation in calculated coefficient ranged from .15 to .30 for starting and from .08 to .23 for running.

LOAD ATTENUATOR RATINGS (8 RODS)

Start F = 4(1163) = 4650 lb. ±6 percent

Run F = 4(1098) = 4390 lb. ±6.4 percent

AvSER CALIBRATION TEST (DYNAMIC)

See Figure 38.

Based on single test of 2-rod unit, pull force = 1250 lb. start and 1160 lb. run.

Since AvSER Dynamic Test Result is within the spread of slow pull rate results, drop test evaluations will be made using above attenuator ratings.

SEAT DROP TEST UH-1B SEAT

Seat Weight	lb.	
Bucket	30.6	
Bucket Armor	56.3	
Misc. (Est.)	3.1	(Without Seat Cushion)
Filler Block	17.0	
Total	107.0 lb.	(Tests Nos. 2 and 3)
Ballast	15.0	
Total	122.0 lb.	(Test No. 1)

Dummy Weight (185 lb.)

Effective weight on seat (80 percent), 148 lb.

Test No. 1

Total Bucket Weight, W = 122 + 148 = 270 lb.

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Tests Nos. 2 and 3

    Total Bucket Weight, $W = 107 + 148 = \underline{255}$ lb.

Inertia Forces to Actuate Attenuators

$$I = \frac{F}{W/g}$$

Test No. 1

    Attenuator Rating

$$F_{start} = 4650 \text{ lb. } \pm 6\%$$

$$F_{run} = 4390 \text{ lb. } \pm 6.4\%$$

$$I = \frac{/\underline{17.3g} \text{ start}/}{/\underline{16.3g} \text{ run}/} \quad \pm 6\%$$

Tests Nos. 2 and 3

$$I = \frac{/18.2g \text{ start}/}{/17.2g \text{ run}/} \quad \pm 6\%$$

Additional inertia force is required to overcome sliding friction on seat support tubes.

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DROP TEST RESULTS

Three drop tests were performed at Aviation Safety Engineering and Research, Phoenix, Arizona. Results of the tests are reported in Reference 13, AvSER Report No. M-66-16. Significant data which are pertinent to load attenuator study are included herein as convenient. Curves of inertia forces versus time from initial contact between drop cage and honeycomb are included in Figures 40, 41, and 42. Accelerometers of particular interest were those positioned for vertical inertia measurement on the drop cage near the seat mount, on the seat bucket, on the dummy pelvis, and on the dummy chest.

It is observed in each case that the inertia readings of the seat, pelvis, and chest are not in phase. Primary initial acceleration peaks occur in order of proximity to the base. In each case, due to the phase lag, the peak chest inertia is concurrent with minimum seat inertia. This condition necessitates high dummy inertia forces in order to balance load attenuator actuation force. It is apparent that seat weight and cushion depth should be minimized.

Test No. 1

It appears that the total of dummy and seat bucket mass should be analyzed as three discrete masses: seat (122 pounds), dummy pelvis (74 pounds), dummy chest (74 pounds).

Since accelerometer readings for each of the three masses are out of phase, they have been combined using a weighted average technique to arrive at a composite inertia for the total mass. See Figure 40. The resultant composite inertia times the total weight of 270 pounds represents the total vertical force between the seat bucket and support.

In Test No. 1 a maximum composite inertia of 33g is indicated before the attenuator strokes. Vertical force = 33(270) = 8900 pounds. With attenuator rating at 4650 pounds (start), there remains 4250 pounds force which must be carried as friction between bucket adapters and support posts, since these are the only remaining points of contact to react the total force. It is then rationalized through study of Figure 40 and cursory analyses of the seat structure that relative rotation of the lower bucket adapters and the support posts resulted in binding between the adapters and posts. The posts had been coated with a dry lubricant which would have allowed without moment interaction approximately 500 pounds friction resistance instead of 4250 pounds. It can only be concluded that the lubricant so nearly filled the close clearance between the posts and adapters that little or no relative rotation was possible and that a high interaction moment was developed. The joint moment would carry from adapter to post through high contact pressure between the I.D. of the adapters and the O.D. of the posts The phenomenon as thus concluded can be further substantiated by observing the stroke time versus seat acceleration on Figure 40, noting that the stroke occurred while the seat acceleration (hence rotation and binding) was minimized at approximately 0.037 second to 0.05 second.

Minimum cumulative inertia of 14.5g is below the attenuator rating range

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(±6 percent) but could probably be explained as release of elastic energy stored in the seat support and cage during system overload while adapter and posts are binding.

Test No. 2 (See Figure 41)

Test results are similar to those of Test No. 1 except that the attenuator stroke appears to start while seat inertia is high. There is still evidence of binding, however, as indicated in the following force summary.

> At Start of Stroke
> Where Composite Inertia = 28g (start) and total dummy
> and bucket weight is 255 pounds (ballast removed)
>
> Total Vertical Force = 28(255) = 7140 pounds
> Attenuator Rating = 4650 pounds (Start)
> Friction Force = 7140 - 4650 = 2490 pounds
>
> During Attenuator Stroke
> Where Cumulative Inertia = 20½g
>
> Total Vertical Force = 20½(255) = 5230 pounds
> Attenuator Rating = 4390 pounds (Run) ±6.4 percent
> Friction Force = 5230 - 4390 = 840 pounds
>
> Near End of Stroke
> Where Cumulative Inertia = 30g
> Total Vertical Force =30(255) = 7650 pounds
> Friction Force = 7650 - 4390 = 3260 pounds

It is again interesting to note that composite inertia varies somewhat in proportion to bucket inertia, thus tending to substantiate the hypothesis (previously implied) that bucket rotation and subsequent binding are generally in phase with bucket inertia force.

The minimum friction force of 840 pounds is within the range of initial predictions for adapter-post friction without binding. Pretest expectation was to realize a constant friction force approximately equal to 10 percent of total vertical force.

Test No. 3 (See Figure 42)

The inertia curve patterns are of similar characteristics as Test No. 2 curves except for peaks of approximately 0.06 second caused by load attenuators' bottoming as load is picked up by stroke limiting safety cables.

The same phenomena as discussed previously under Tests Nos. 1 and 2 are exhibited in varying degree.
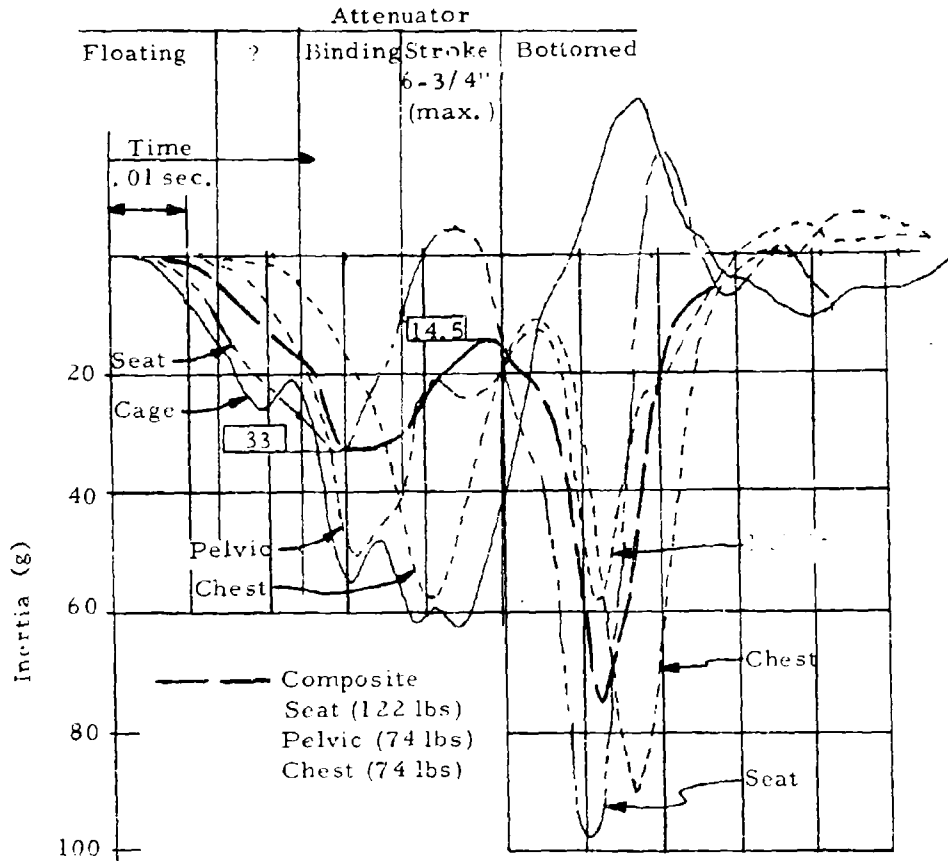
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Figure 40. UH-1B Seat Drop Test Results, Test No. 1.
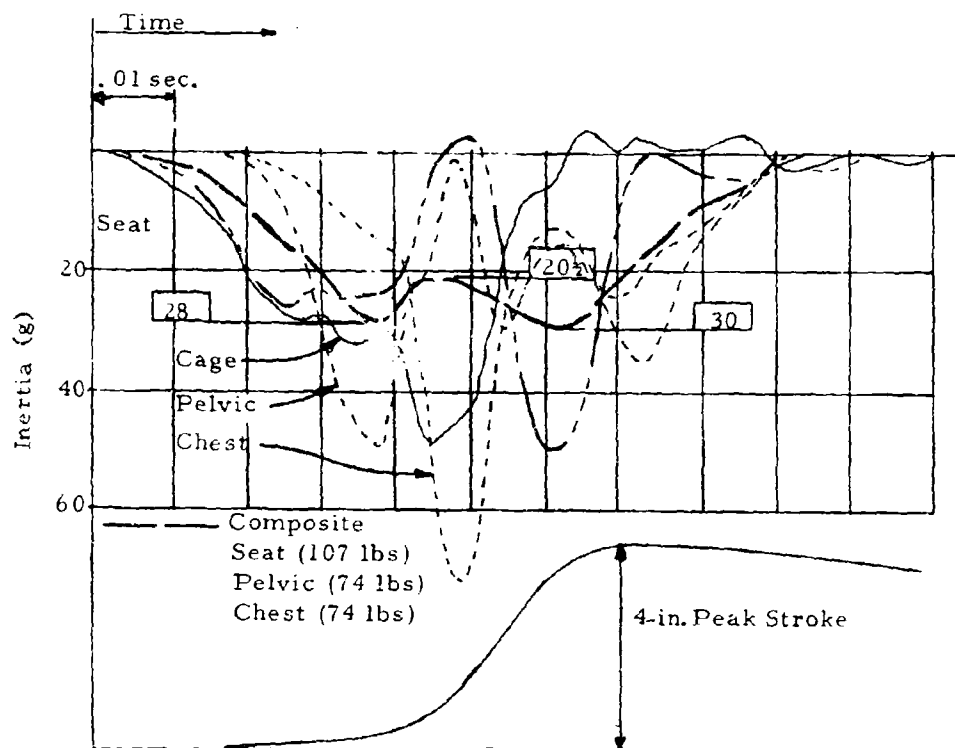
Figure 41. UH-1B Seat Drop Test Results, Test No. 2.

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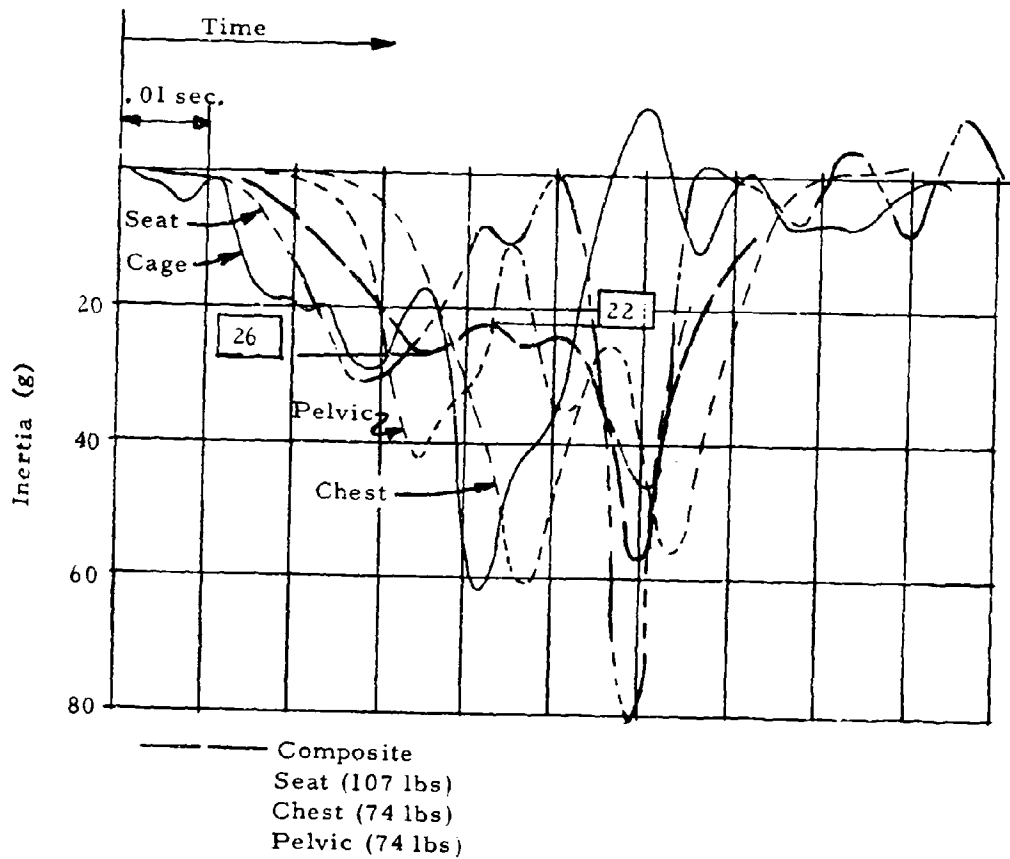
Figure 42. UH-1B Seat Drop Test Results, Test No. 3.

SUMMARY OF ENERGY ABSORPTION STUDY

It has been noted in the analyses of the drop tests at AvSER that the seat bucket adapters and support posts on the UH-1B seat tend to rotate in opposite directions; consequently, binding between the adapters and posts caused very high friction force to resist vertical motion of the seat bucket. This condition was probably inadvertently aggravated by application of dry lubricant to the support posts because the lubricant further restricted free relative rotation between the adapters and posts by nearly filling the adapter-post clearance. In one case, Test No. 1, the total vertical inertia force of the bucket-dummy composite mass was almost twice the attenuator rating of 4650 pounds start. Obviously this condition has negated clear, concise interpretation of force carried by the load attenuators. The preliminary development tests, however, do verify that the attenuators actuate with an essentially constant force for the entire stroke. See Figures 38 and 39. The spread in preliminary test pull values of ±6 percent could be virtually eliminated by closer tolerance on finish of moving parts, by lubrication, or by spindle bearings or sleeves. Further developmental refinement could be accomplished to reduce friction and thereby optimize the attenuator concepts within closer tolerance on rated pull force.

It should be observed or the Hayes seat design that binding such as experienced in drop tests of the UH-1B seat cannot occur because the adapters (fittings) which slide n the support posts are not connected directly to the seat bucket and consequently are free to rotate with the posts except for slight frictional resistance. Accurate evaluation of attenuator loadings on the Hayes seat should be assured.

The attenuators shown on the Hayes seat design are sized analytically to accommodate the calculated inertia forces within contract requirements and as summarized on Table VIII. Summary of attenuator sizes and ratings for the Hayes seat design is given in Table XIV. The sizes as tabulated may be subject to slight modification pending results of developmental testing.

TABLE XIV
SUMMARY OF ATTENUATORS
PRELIMINARY SIZING

	With Armor		Without Armor	
	200-lb. Occupant	135-lb. Occupant	200-lb. Occupant	135-lb. Occupant
	Desired Attenuation Force*			
Vertical (lb.)	4960	4355	3360	2630
Inertia (g)	17	17	17	17
Lateral Upper (lb.)	1400	1493	1276	1345
Lateral Lower (lb.)	1537	1547	1473	1472
Inertia (g)	11.15	13.6	16.0	21.6
Forward (lb.)	8505	8505	6160	6160
Inertia (g)	22.5	25.9	22.5	29.1

Preliminary Attenuator Sizing with 4130 Steel Rods, 150 Ksi Ultimate

Vertical (lb.)	(8 Units)	(7 Units)	(5 Units)	(4 Units)
	(5000)	(4375)	(3125)	(2500)

Total of 8 Units with '25-lb. rating: Single Spindle, Rod Dia. = 5/32 in., Spindle Di . = .493 in.

Lateral Upper - 2 Units with '00-lb. rating: Single Spindle, Rod Dia. = 3/16 in., S; dle Dia. = .823 in.

Lateral Lower - 2 Units with 750-lb. rating: Single Spindle, Rod Dia. = 3/16 in., Spindle Dia. = .750 in.

Forward (Armor) - 4 Units with 2125-lb. rating: Double Spindle, Rod Dia. = 7/32 in., Spindle Dia. = .845 in.

Forward (No Armor) - 4 Units with 1540-lb. rating: Double Spindle, Rod Dia. = 3/16 in., Spindle Dia. = .725 in.

* Reference Table VIII.

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4. DESCRIPTIVE NOTES (Type of report and inclusive dates)

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11 SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY
	US Army Aviation Materiel Laboratories
	Fort Eustis, Virginia

13 ABSTRACT The primary objective of this program was to develop the design of an improved aircrew armored crash survival seat. The U.S. Army, through previous contractual efforts, designed and developed four armored aircrew seats having crash load attenuation features. Each design was unique in concept, geometry, material, energy absorption and other factors. A comprehensive analysis and evaluation was made of these designs as a basis for developing an improved design.

A seat design was developed that will meet or exceed the specified requirements. It is forward facing and designed for installation in the UH1B aircraft. It consists of a seat bucket, fabricated of aluminum alloy sheet, supported by a tubular steel framework from floor tracks. Load attenuation is accomplished by a Hayes-developed energy absorbing device utilizing controlled bending of steel rods. The seat bucket is allowed to move relative to the support for energy absorbing stroke in the vertical and lateral directions. The complete seat moves relative to the floor for the stroke in the longitudinal direction. The load attenuation devices can be adjusted or easily replaced for various occupant-armor weight configurations. Vertical adjustment is accomplished with an electric actuator.

Ballistic protection is provided by a shell of ceramic-fiber glass composite armor attached to the seat bucket. The armor was developed by CTL and is somewhat lighter in weight than present armor. Protection is provided against 7.62 mm APM-61 ammunition fired at 100 yards range at 15° obliquity.

DD FORM 1473

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Optimized Aircrew Seat Design Study						
Attenuating Devices in Vertical, Longitudinal, and Lateral Planes						

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